A LOW ENERGY LAKE DESTRATIFIER

By

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CHAPTER I

INTRODUCTION

The Problem

Water supply for human use comes mostly from ground water or from water impounded on the surface. The storage of water in large lakes leads to substantial problems of a physical, chemical and biological nature. Impounded waters experience certain levels of degradation in quality. Water quality has almost as many meanings to individuals as has water resources itself (1). Most commonly attention is directed to the pollution of natural or man-made water storage by the acts of man and by natural causes. Summer stratification in reservoirs creates large volumes of poor quality water below the thermocline. Thermal stratification occurs in practically all reservoir impoundments (2). Conventional structures in most dams withdraw low-flow releases and power releases from the hypolimnion resulting in serious degradation in long reaches of streams below the dams.

Cold water released from the bottom of Shasta Reservoir in California has reduced crop yields on irrigated land (3). When the bottom water of a stratified reservoir is used as a source of drinking water, an additional load is placed on the water treatment plant. Hypolimnion releases often cause downstream fish kills and are irritating to people in the adjacent area due to the presence of toxic gases that are released by the water.

In recent years (4), as the demand for water has approached in magnitude the available supply, there has been a profound expansion of the goals of agencies dedicated to water quality control, such as protection of the public health, aesthetics, or social goals of water quality.

Engineers and Biologists have been concerned with the adverse effects of stratification. As a result of their interest, a great deal of work has been done and includes:

a) Use of mechanical pumping to investigate the effect of pumping cold water from the bottom of a stratified water body to be discharged at the surface.

b) Use of pure oxygen for a means of artificial aeration.

c) Pumping of compressed air to increase aeration of the body of water as well as to induce vertical movement of the water to break that stratification; but mainly, to bring bottom water to the surface so oxygen can be transferred to it from the atmosphere. Compressed air has been used more than any other method to raise water from the bottom to the top of lakes.

Symons (5) shows that artificial destratification, although not the only engineering technique for raw-water quality control, improves water quality in many ways, thereby reduced burdens on water treatment plants and does not significantly worsen water quality in any way. He specifically concludes that bringing hypolimnion water to the surface of a reservoir with a mechanical or diffuser-air pump eliminated thermal stratification.

A low energy lake destratifier is proposed in this research. It consists of a pump with a propeller that produces a movement of the water from the top to the bottom of the body of water. The objective is to pump a large flow of water with a very low input of energy. The propeller is located about four feet below the surface of the lake, and the water moves downward and is expected to prevent stratification and mix the water of the lake.

Objectives

The objectives of this study are:

1. To design, construct and test a pump that will pump large amounts of flow with low input of energy.

2. To determine the relationships of the flow rate through the pump to RPM, diffuser skirt diameter and diffuser length.

3. To determine the relationships of horsepower to RPM, as other variables are varied.

4. To calculate the expected head loss through the device, based on available coefficients and measured velocities.

5. To make estimates of pump efficiency from measured values of horsepower and flow rate using calculated values of head.

6. Using the laws of similitude, to establish dimensionless parameters and relationships to assist in the design of larger models.

Limitations of the Study

Diffuser lengths were limited by lake depth to a length of 24 feet or less; because of the short skirt, a maximum outlet diameter of eight feet was studied.

This study was limited to a flexible skirt of nylon-reinforced neoprene.

The maximum propeller RPM, and flow rates Q, were limited by the 1/2 horsepower electric motor used in the study.

The study was limited to a development of equipment for moving large volumes of water with low power and was not a study of the effects of these flows on such stratification parameters as temperature, dissolved oxygen or biological effects.

CHAPTER II

REVIEW OF LITERATURE

The presentation of this review is not intended to be a comprehensive review of the subject. The author has selected the areas to be most applicable to this problem for review. The derivation of basic laws or formulae concerning pumps are not reproduced here. Instead the reader is referred to the references for derivation of the equations.

The presence of destratification in water stored in ponds and lakes has been the concern of many people, since thermal or density stratification causes a general deterioration of water quality in the deeper areas of the impoundment.

Stratification is widespread, occurring at all latitudes in the United States (6). Not only thermal, but a chemical stratification takes place during spring, summer and fall seasons.

During summer, especially, stratification is present in three well defined stratas: epilimnion (Upper layer), hypolimnion (Bottom layer) and a layer of discontinuity called the thermocline. (Figures 1, 2 and 3)

Leach (7) indicated that as the epilimnion warms, the thermocline is developed and acts as a diaphragm. This situation prevents surface-induced circulation below that depth.

Thermal stratification results in deterioration of water for both

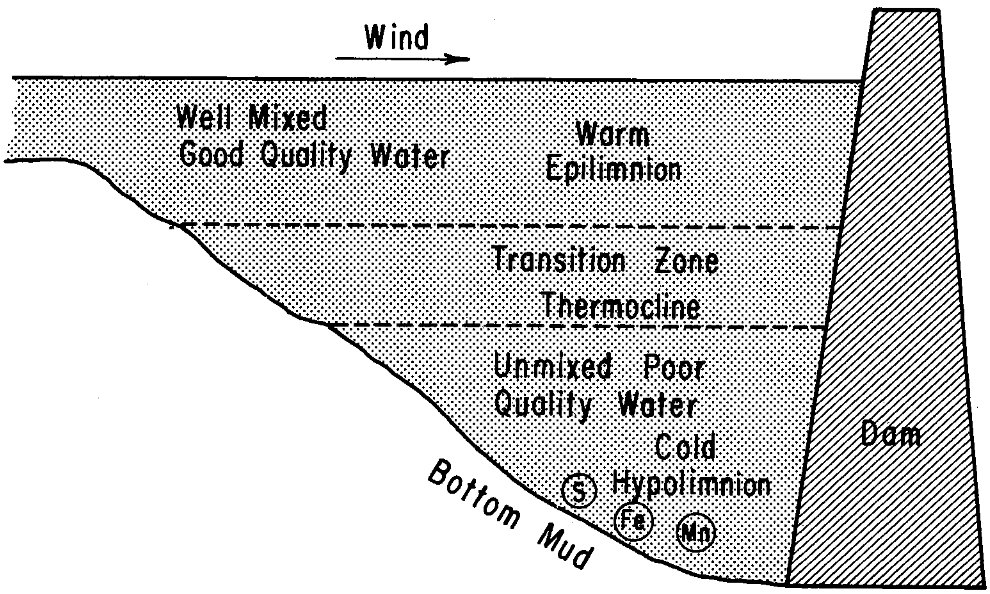

Figure 1. Typical Summer Thermal Stratification Pattern
After Symons, J. M. (5)

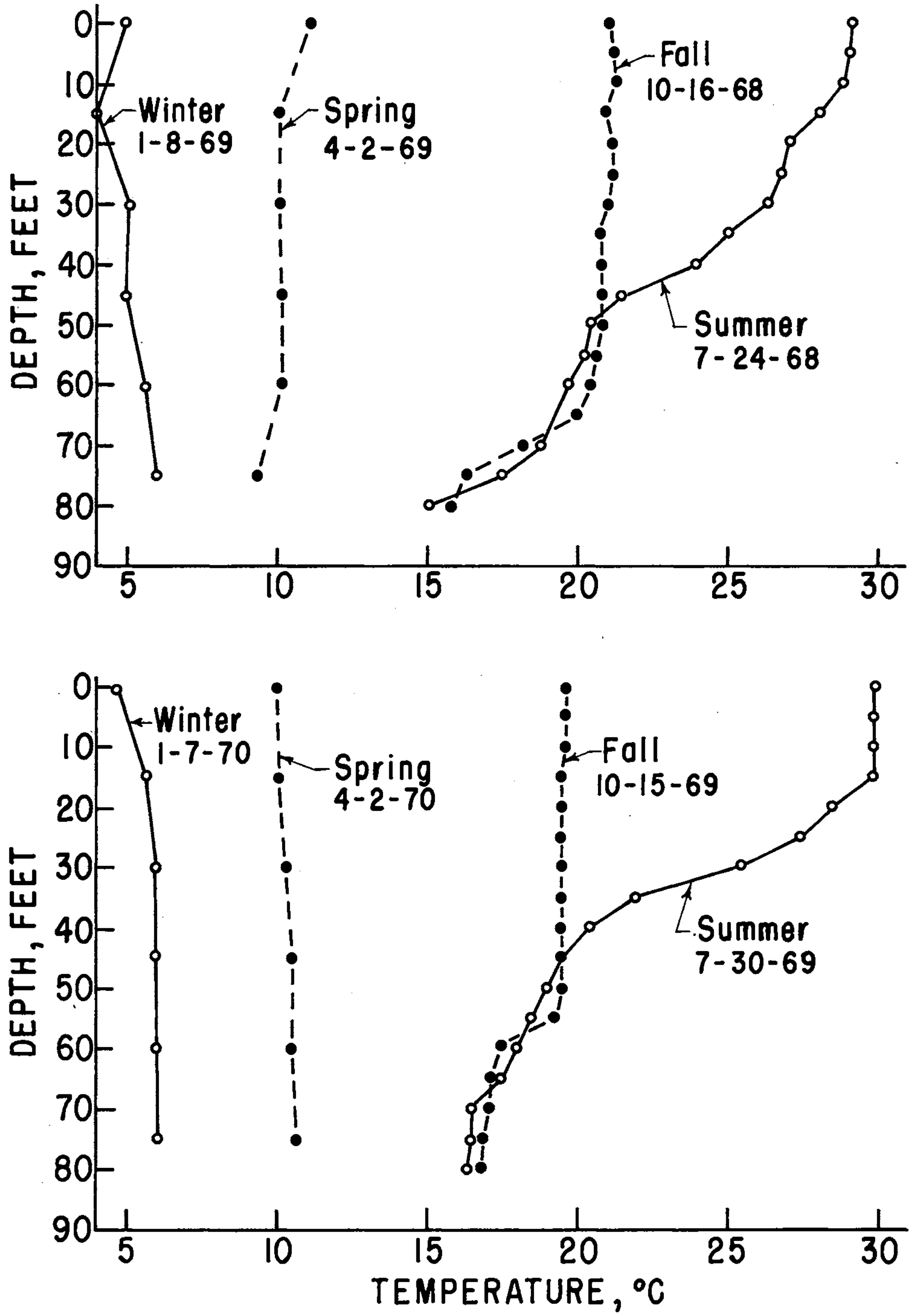

Figure 2. Temperatures for Selected Dates at One Place at Lake Arbuckle, Oklahoma. After Duffer, W. R. and C. C. Harlin, Jr. (8)

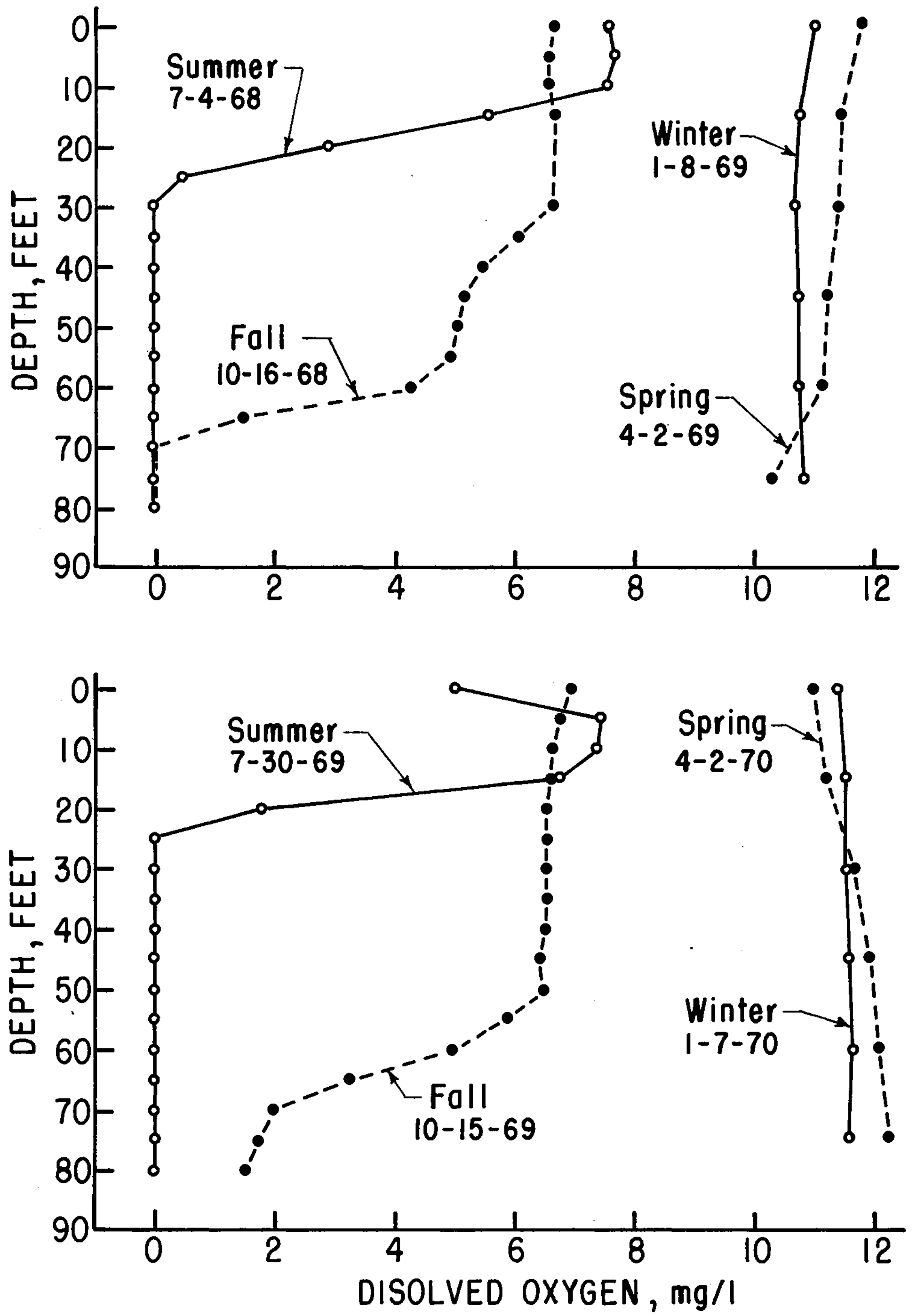

Figure 3. Dissolved Oxygen for Selected Dates at one Place at Lake Arbuckle, Oklahoma. After Duffer, W. R. and C. C. Harlin, Jr. (8)

human and animal use. Fast (9) stated that thermal stratification of eutropic lakes often results in deterioration of drinking water quality, anaerobic and corrosive conditions, increased evaporation rates, reduced heat budgets and other undesirable properties within the lake.

With the existing research, it is possible to make predictions of the behavior of an impoundment as well as its variations concerning its internal temperatures. Markofsky (2) indicated that the thermal stratification through the density variation, has a predominant influence on the flow pattern and circulation within the reservoir.

The amount of oxygen dissolved in a reservoir is of paramount importance. It is the result of the input of oxygen such as the contribution from the atmosphere, photosynthesis and aquatic life.

This balance, as stated by Markofsky, is dependent on numerous physical and biological factors which include convective transport by internal currents, atmospheric reaeration at the surface, photosynthetic oxygen sources associated with plant life, oxygen demands of river inflows, bottom deposits and respiration and decomposition of aquatic organisms.

The reasons for destratifying an impoundment are (10,11,12):

a) Improvement of water for beneficial use.

b) Mixing hypolimnion water with the epilimnion to prevent anaerobic conditions in the bottom waters.

c) Diminishing algae blooms.

d) Reduction of taste and odors attributed to products of anaerobic decomposition by oxidation to less obnoxious forms.

e) Breaking up the thermal stratification.

Methods of Artificial Destratification

Reaeration of reservoirs has been primarily concerned with mixing by diffused air or pumping with accompanying atmospheric reaeration at the water surface.

There are two broad classifications of systems to create thermal destratification. They are mechanical pumping, and compressed air or oxygen releases near the bottom of the impoundments.

Mechanical Pumping

It is known that artificial aeration can be applied successfully to raise dissolved oxygen using mechanical surface aerators (13). Some types of aerators agitate water in fairly large quantities to increase the oxygen content at the surface.

Hooper (14) reported on the earlier works done to break stratification by mixing the impounded water with a pump, and his results were characterized as very slow with a low efficiency. He pumped the cold water from the bottom to the top of the lake.

Symons (5) used a system in which he induced pumping to the surface of a lake. The work required was that needed to lift the differences in water density, overcome the inertia of standing water, create a velocity and overcome the hydraulic head losses in the mechanical systems. He used a kind of axial-flow type pump, with a pipe attached below the impeller. The unit was mounted on a float. For the twelve inch diameter mixed-flow pump, he used a 16 Hp gasoline engine. The output of the engine was 12 Hp with a pump capacity of 13 Ac.Ft./Day or 2,880 GPM. The corresponding value for velocity of the water discharged from the pump was 8.4 fps.

J.B.F. Scientific Corp (13) describes numerous commercial manufacturers as well as equipment for artificial destratification. The New Jersey Water Resources Research Institute (15) reported the use of a commercial mechanical aerator for surface operation that required 75 Hp. It had 35 vertical steel blades working at about 30 revolutions per minute. The researchers reinforced previous conclusions that mechanical aerators were capable of raising oxygen levels (5,6,7,9).

Compressed Air

A different way to destratify impounded water is to release compressed air or gaseous oxygen. The air creates a vertical movement of water so that aeration is induced in the hypolimnion strata. The application of gaseous oxygen as well as the application of compressed air to a body of impounded water in order to get an imput of oxygen to improve biological reactions has been used in waste treatment plants (16).

Leach (7) reported the use of compressed air in destratification of a very large reservoir, Eufaula reservoir in Eastern Oklahoma. The application of air was through diffusers that were located below the water surface; the air being supplied by compressors.

Bernhardt (17) reported the use of a duct 6.6 ft. in diameter, with a length of 70 ft. that produced a flow rate of 67.9 cfs of water when air was supplied by a 40 Hp compressor.

A combination of systems is also possible, consisting of bringing water from the bottom to be discharged at the surface where a mechanical aerator produces the aeration (16).

The application of a gas transfer process in aeration has been described by Speece (18) as the U-tube aeration system. Air bubbles are injected into a water flow that follows a U configuration. The aerated water is returned to the hypolimnion.

Bernhardt (17) reported complete destratification using a diffused air bubble method to aerate the hypolimnion, preserving a quantity of cold water for use as drinking water. A compressor of 36.5 kw was used.

Several mechanical pumping systems have been devised to establish circulation and create destratification of large bodies of water. However, attention has not been directed toward improving the input of energy requirements. Most of the equipment described by the present available literature has high power requirements (11,12,17,19,20). One exception is a system presented by Speece (21) in which he predicts the possibility of moving 4,500 gpm with one horsepower.

Closed Conduit Flow

Closed conduit flow occurs in a closed conduit, at full capacity and under pressure.

An analysis of this fluid flow problem requires the application of the equation of continuity, the energy principle and the principles and equations of fluid resistance.

The concept of the equation of continuity is shown in Figure 4, where 1 to 2 represents the section under consideration. The cross sectional areas, are A_1 and A_2; the mean fluid densities are ρ_1 and ρ_2; and the velocities in each section, represented by V_1 and V_2;

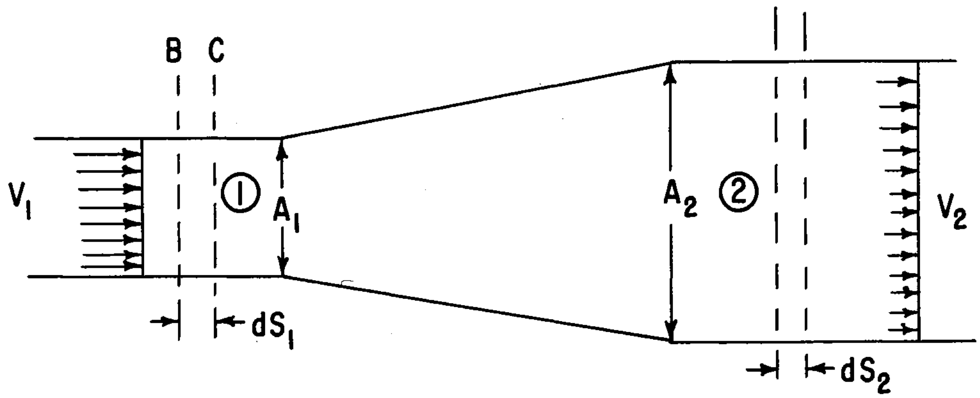

Figure 4. Description of the Continuity Equation in Closed Conduit Flow

The equation of continuity results from the principle of conservation of mass. For steady flow, the mass of fluid passing all sections in a stream of fluid per unit of time is the same. For an incompressible fluid the equation is expressed as follows:

$$Q = A_1 \times V_1 = A_2 \times V_2 \tag{2-1}$$

where:

Q = Flow per unit of time, fps

In Figure 5, Bernoulli's equation applies the theory of conservation of energy. The total energy at section 1 at a point which has negligible velocity is equal to the energy at section 3 plus losses. The equation for incompressible fluid motion is:

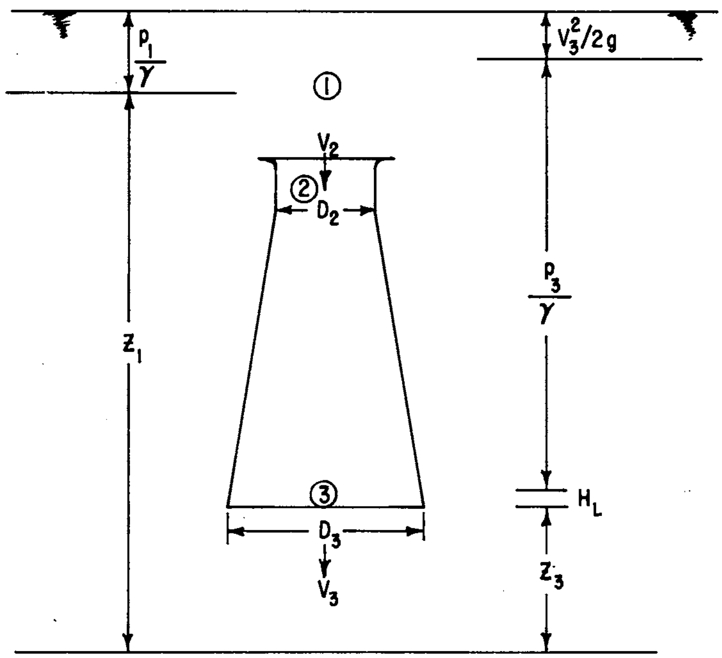

Figure 5. Description of the Total Energy Relationships in the Pump

$$\frac{p_1}{\gamma} + \alpha_1 \frac{(V_1)^2}{2g} + Z_1 = \frac{p_3}{\gamma} + \alpha_3 \frac{(V_3)^2}{2g} + Z_3 + H_L \qquad (2\text{-}2)$$

where:

γ = Specific weight, pounds per cubic foot

p_1 = Pressure intensity at section 1, pounds per square foot

p_3 = Pressure intensity at section 3, pounds per square foot

V_1 = Velocity at section 1, fps (Assumed zero)

V_3 = Velocity at section 3, fps

Z_1 = Elevation above datum, ft.

Z_3 = Elevation above datum, ft.

H_L = Total head loss, ft.

g = Acceleration due to gravity, ft./sec^2

$(V_3)^2/2g$= Exit velocity head, ft.

p/γ = Pressure head, ft.

Z = Position due to differences in elevation, ft.

α_1 and

α_3 = Energy coefficients, assumed to be 1.0

Having H_T representing the loss of head due to all causes, Bernoulli's equation (2-2) can be expressed as:

$$\frac{p_1}{\gamma} + Z_1 = \frac{p_3}{\gamma} + \frac{V_3^2}{2g} + Z_3 + \Sigma H_L \qquad (2\text{-}3)$$

$$H_L = H_1 + H_2 \qquad (2\text{-}4)$$

$$H_T = H_L + \frac{(V_3)^2}{2g} \qquad (2\text{-}5)$$

$$H_T = H_1 + H_2 + H_3 \qquad (2\text{-}6)$$

where:

H_1 = Loss of head at entrance that occurs when the water enters the conduit from a comparatively large body of quite water, ft.

H_2 = Loss of head due to friction and fluid turbulence in the expanding jet, ft.

H_3 = Loss of head at the exit, when the conduit discharges into a large body of quiet water, ft.

H_T = Total dynamic head of the pump, ft.

Determination of Losses

Loss of Head at the Entrance

A flow acceleration takes place at any type of entrance, creating a head loss. Hamilton (22) presents the loss as:

$$H_1 = K_1 \frac{(V_2)^2}{2g} \tag{2-7}$$

where:

K_1 = Entrance loss coefficient

In the present study a bell-mouth entrance is used, and a value of 0.10 for K_1 is indicated (22). The value $(V_2)^2/2g$ is the velocity head in the throat.

Loss of Head Due to Enlargement

Gibson (23) determined that the loss of head due to gradual enlargement is intimately related to the shape of such enlargement. Considering section 2 and section 3, the expression is:

$$H_2 = K_2 \frac{(V_2 - V_3)^2}{2g} \qquad (2\text{-}8)$$

where:

H_2 = Loss of head due to enlargement, ft.

K_2 = Loss coefficient for sudden expansion, ft.

V_2 = Velocity before the widening commences, fps

V_3 = Velocity after the widening has ceased, fps

The coefficient K_2 is formed by the combination of wall friction effects and large scale turbulence.

It may be observed that the lowest value for the head loss is found when the angle is between five degrees and six degrees, having a value of K_2 near 0.135 (Figure 6). An increase in the angle increases the theoretical loss of head for the angles larger than six degrees and smaller than seventy degrees.

Vennard (24) when analyzing the values of Figure 6 concluded that the pressure rise through the diffuser, computed from the application of the Bernoulli's equation, will be larger than that which actually can be realized.

Loss of Head at Exit

As the discharge enters the reservoir, the dissipation of the velocity head in the leaving water causes a loss estimated as:

$$H_3 = K_3 \frac{(V_3)^2}{2g} \qquad (2\text{-}9)$$

where:

H_3 = Loss of head due to exit, ft.

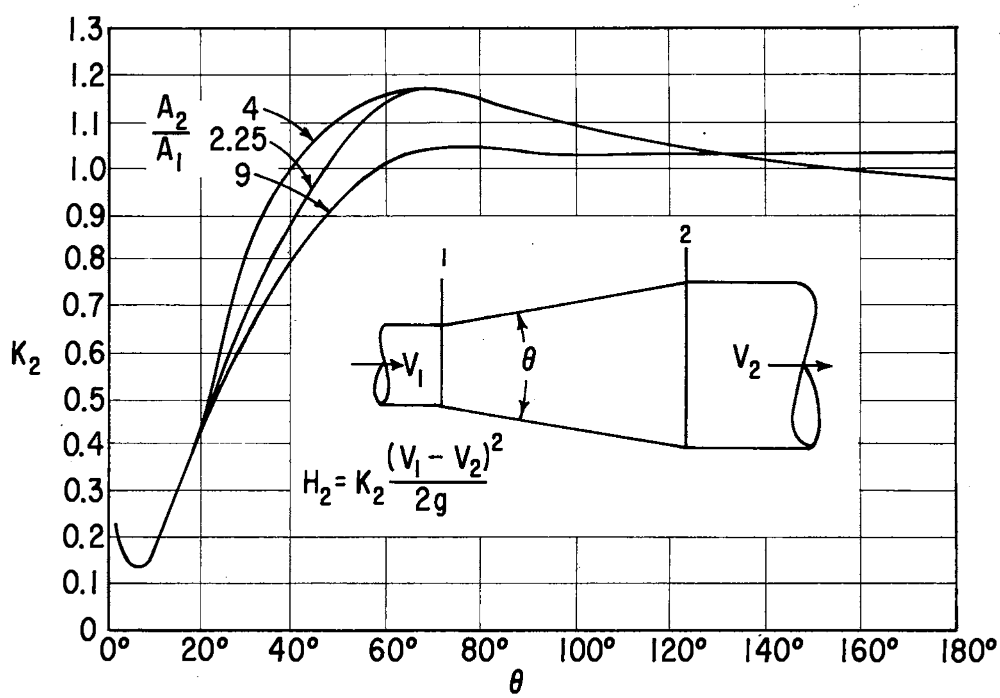

Figure 6. Loss Coefficients for Conical Enlargements (24)

V_3 = Velocity at the outlet section, fps

K_3 = Exit loss coefficient

Based on the assumption that K_3 = 1.0 (25), the entire kinetic energy of the water entering the reservoir is assumed to be converted into heat energy by the turbulent mixing that takes place.

Production of Fluid Flow

There are six methods by which fluids can be made to flow through a conduit or channel (26): a) By action of centrifugal force; b) By volumetric displacement, accomplished mechanically or with other fluids; c) By mechanical impulse; d) By transfer of momentum from another fluid; e) By the use of electromagnetic force, and f) By gravity. Regardless of the physical characteristics of the fluid, whether it is compressible or incompressible, these six methods include all the available means of fluid transport.

Pumping Units

The mechanical devices for establishing pressure to create a flow through a closed conduit may be classified as: a) Centrifugal pumps, b) Rotary Pumps, and c) Reciprocating pumps. Only centrifugal pumps are considered in the present review.

Centrifugal Pumps

A centrifugal pump consists primarily of an impeller and a stationary casing. The impeller, imparts the velocity to the fluid pumped. The casing guides the fluid to and from the impeller and converts the velocity head to pressure head. Fluid is supplied to the rotating impeller at or near its center. The various classes of these pumps are defined according to the impeller design which varies from radial to axial flow types.

Radial Flow Pumps

A radial flow pump is a pump in which the impeller directs the flow of fluid by centrifugal force, radially to the periphery of the impeller. The velocity head is largely converted to pressure head in the discharge diffuser.

Mixed Flow Pumps

A mixed flow pump is a pump in which the head of the fluid is developed as a combined effort with a thin impeller, partly by use of centrifugal force, and partly by the push of the vanes. This is accomplished by making the vanes doubly curved or screw shaped so that the discharge is a combination of axial and radial flows.

Axial Flow or Propeller Type Pumps

An axial flow pump, which was the type used in this study is a pump in which most of the head produced by the propeller is due to the pushing or lifting section of the vanes. The fluid enters and leaves the impeller in an axial direction. It operates just like a venting fan enclosed in a tube, except that it moves liquid instead of air (Figure 7). Figure 8 shows a plot of characteristics of a typical propeller pump.

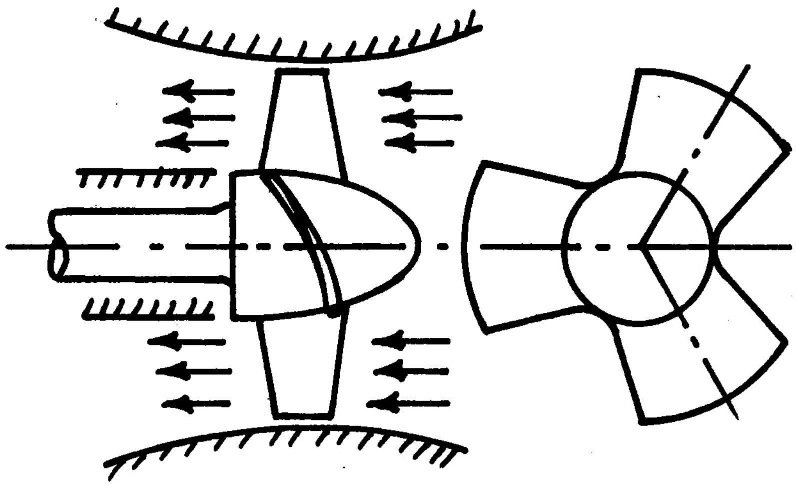

Figure 7. Axial Flow or Propeller Type Pump

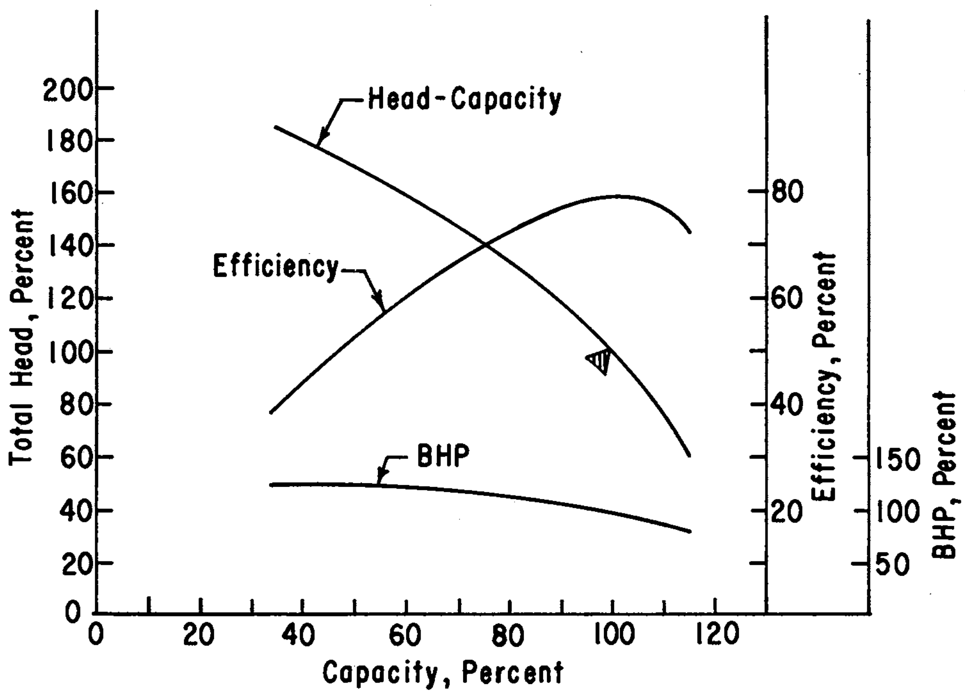

Figure 8. Plot of Characteristics of a Typical Propeller Pump (27)

Pump Performance and Characteristics

The performance of a centrifugal pump is described by the rate of flow, Q; the head of the fluid pumped, H; the power input, P; and the speed of rotation of the propeller, N. Where:

Q = Rate of flow, gpm.

H = Head of fluid, ft.

P = Power input, Hp.

N = Speed of rotation of propeller, rpm.

Affinity Laws

The flow and the other factors are governed by the laws called affinity laws in which it is said: a) Pump capacity increases directly with speed, b) Pump head increases directly with the square of the speed, and c) Pump horsepower increases directly with the cube of the speed. Wislicenus (28) presents these concepts for the analysis of similar pumps:

$$Q_1/Q_2 = N_1/N_2 \qquad (2\text{-}10)$$

$$H_1/H_2 = (Q_1)^2/(Q_2)^2 = (N_1)^2/(N_2)^2 \qquad (2\text{-}11)$$

$$P_1/P_2 = (N_1)^3/(N_2)^3 \qquad (2\text{-}12)$$

Specific Speed

The above laws lead to the presentation of the specific speed concept. The specific speed of a centrifugal pump is defined by the relationship:

$$Ns = N\sqrt{Q}/H^{3/4} \qquad (2\text{-}13)$$

where:

Ns = Specific speed.

N = Speed of rotation of the propeller, rpm.

Q = Rate of flow, cfs.

H = Head of fluid, ft.

Specific speed relates the capacity and head of a pump at its most efficient point. The specific speed varies from zero to infinity for a given impeller and has a constant value for different speeds and sizes of impellers operated at the same efficiency. The specific speed has no physical meaning and is merely a convenient number used to characterize pumps with geometrically similar impellers. Variations of specific speed, Ns, leads to classification of head discharge characteristics (29). Low values of specific speed, say values near 500, are characteristic of high head and small discharge pumps. Intermediate values near 2,000 for intermediate head and discharge pumps, and high values near 10,000 or more are characteristic of low head and large discharge pumps. Radial flow pumps are included in the first group, mixed flow pumps in the second group and propeller flow or axial flow correspond to the third group.

Efficiency of Centrifugal Pumps

Efficiency is a function of impeller design, flow rate and viscosity. Figure 9 shows the relationship of specific speed, and efficiency. The two factors are interrelated and are general indices of design and operating characteristics. As work corresponds to the power expended in a certain length of time, the expression water horsepower is defined as the power theoretically required to lift a

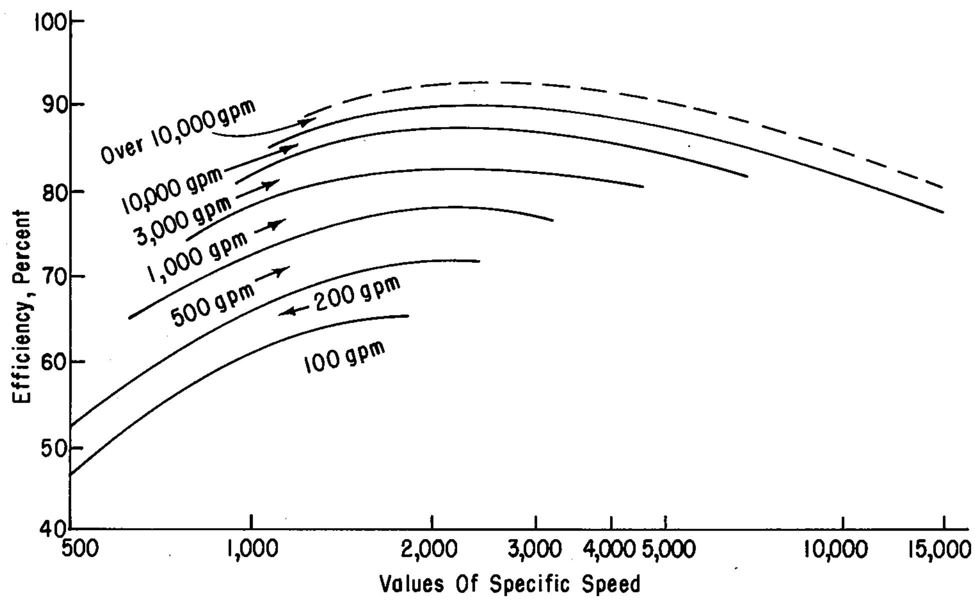

Figure 9. Relationships of Efficiency and Specific Speed in Pumps

given quantity of water each unit of time to a specific height; thus:

$$\text{W.H.P.} = \frac{\text{Lbs. of liquid/min} \times \text{Total dynamic head (ft.)}}{33000} \qquad (2\text{-}14)$$

or

$$\text{W.H.P.} = \frac{Q \times H \times S}{3960} \qquad (2\text{-}15)$$

where:

Q = Flow of liquid, gpm.

H = Total dynamic head, ft.

S = Specific gravity of pumped liquid, taking 1.0 for water.

W.H.P. = Water horsepower, Hp.

When expressing the ratio between energy output to energy input, the value of efficiency can be determined as:

$$\text{Efficiency (Eff)} = \frac{\text{Output}}{\text{Input}} = \frac{\text{W.H.P.}}{\text{B.H.P.}} \qquad (2\text{-}16)$$

where:

B.H.P. = Brake Horsepower, Hp.

By combining equations (2-15) and (2-16) a general equation is obtained:

$$\text{Hp} = \frac{Q \times H}{3960 \times \text{Eff}} \qquad (2\text{-}17)$$

CHAPTER III

EXPERIMENTAL EQUIPMENT

Destratifier Unit

Pump

The axial flow type of pump consisted of a propeller, a stationary casing and a diffuser (Figures 10 and 11). The seven-bladed propeller had an effective diameter of 41-3/4 inches. The blades were 15 inches long and had a width of 5-1/2 inches at the tip and 4 inches at the hub. The angle of the blade was 20 degrees at the base and 10 degrees at the tip (Figures 12 and 13).

The pump body or casing consisted of the air fan housing shroud with a bell-mouth type entrance having a radius of curvature of 2-3/4 inches. A cylinder was added, as shown in Figure 14. The total length was 29 inches and the inside diameter was 42 inches. The width of the circular ring on top was 12 inches.

Under the pump, a plastic diffuser or skirt was installed (Figure 15). This diffuser consisted of a cone-shape tube with its largest size having the following dimensions:

Throat diameter	3.5 ft.
Outlet diameter	8.0 ft.
Length	24.0 ft.

In accordance with the experimental design, the size of the diffuser

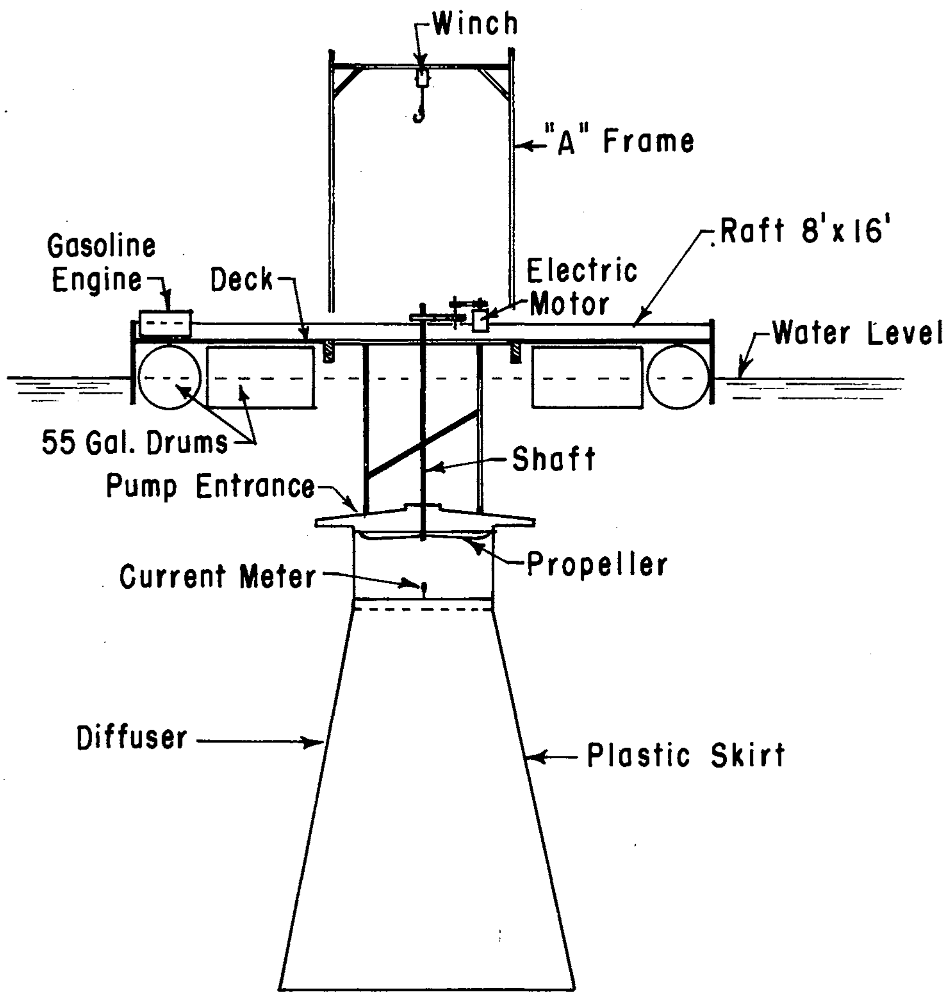

Figure 10. Assembly of Pump and Raft

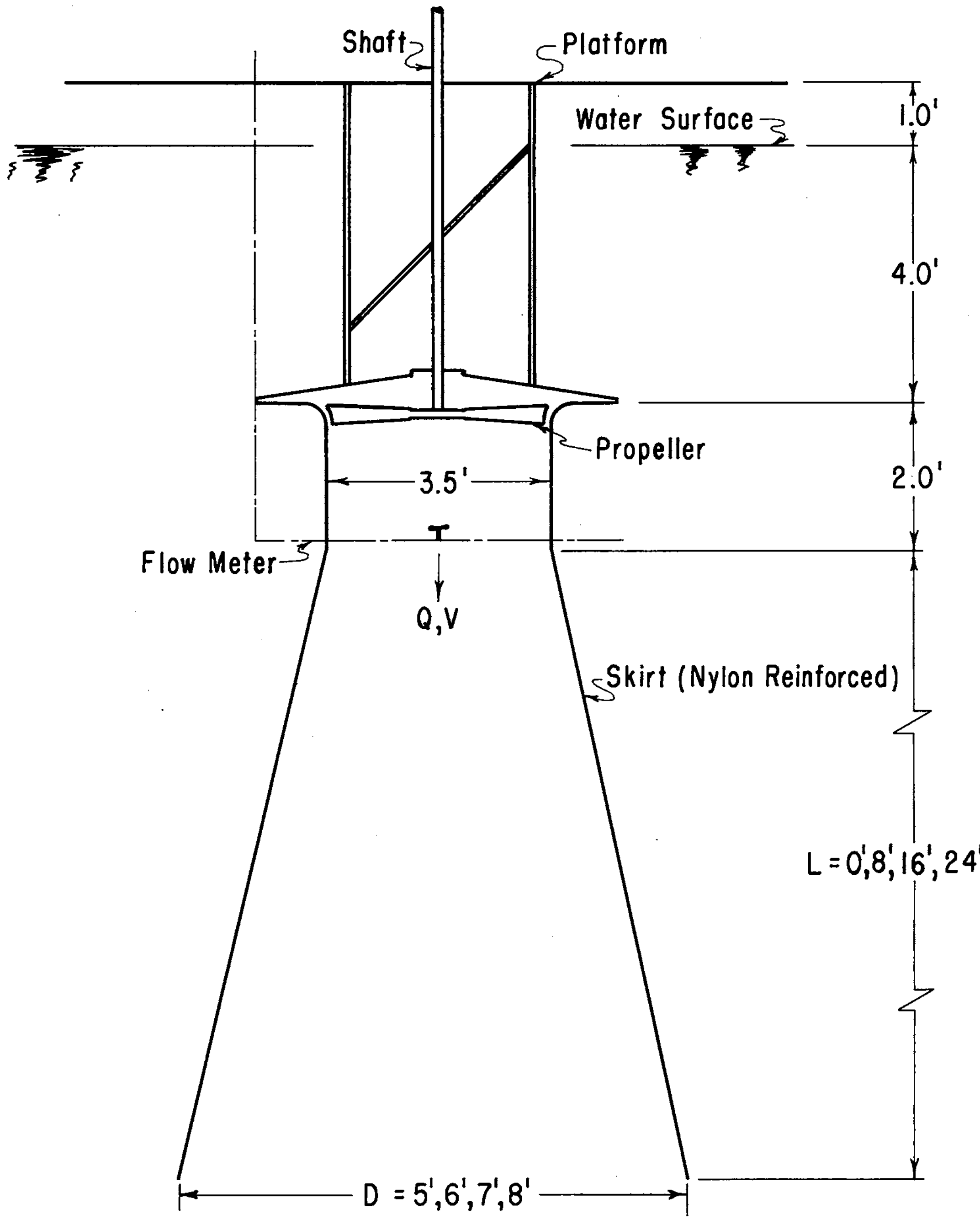

Figure 11. Pump Model

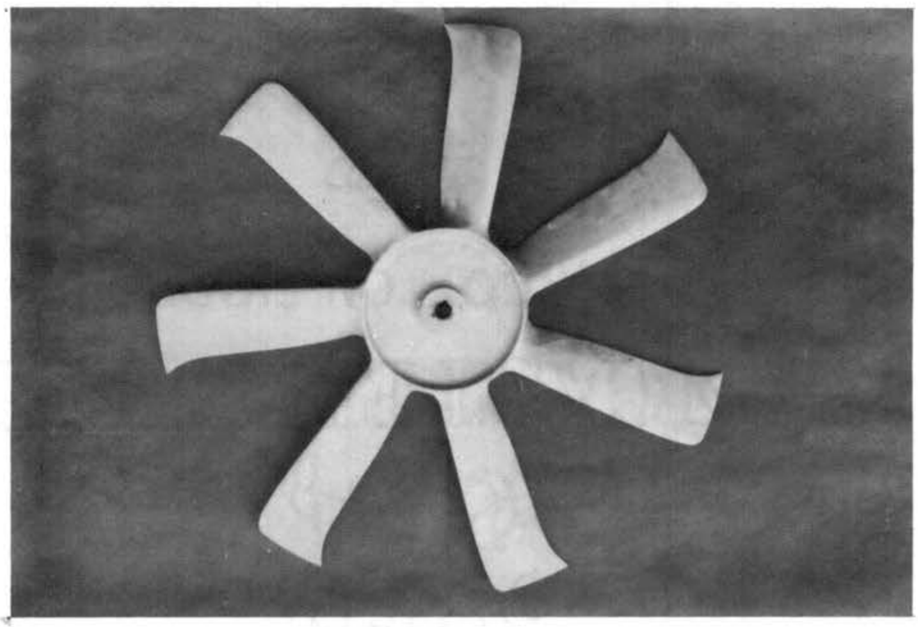

Figure 12. Seven Bladed Propeller Used

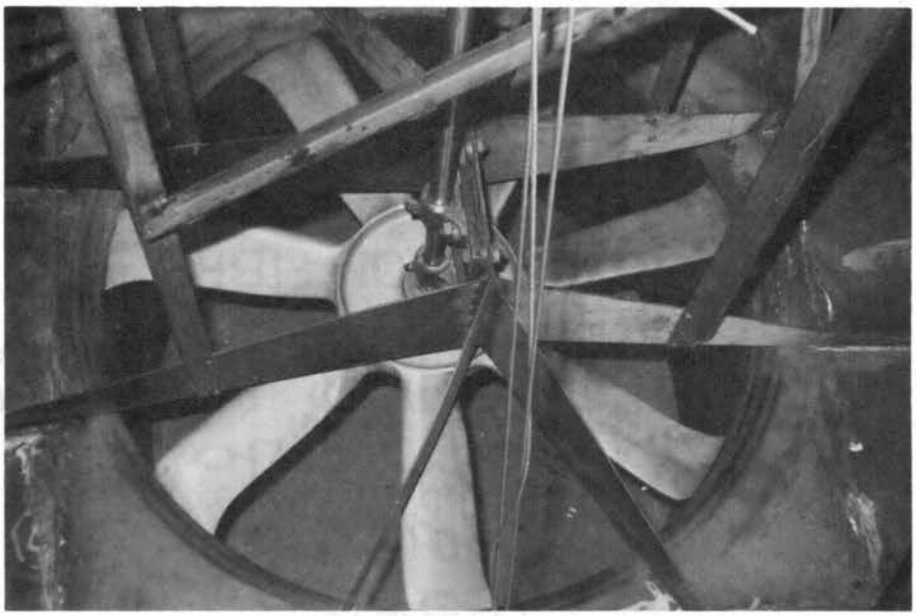

Figure 13. Assembly of Propellers and Casing

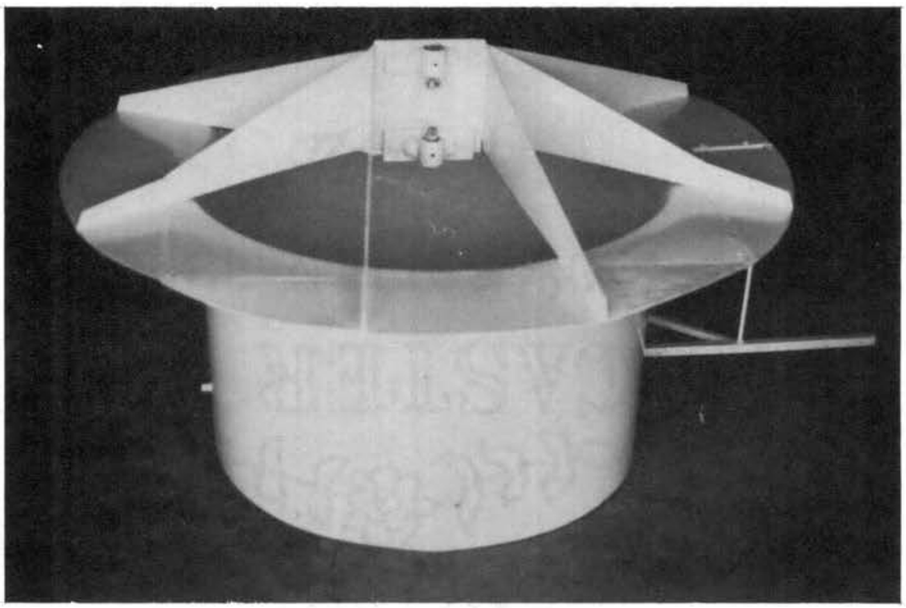

Figure 14. View of Casing of Pump

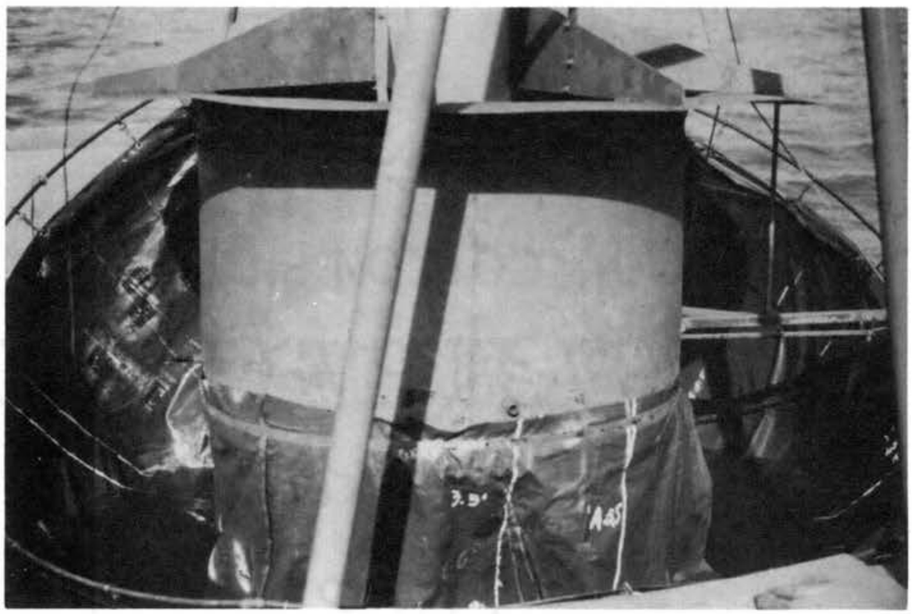

Figure 15. View of Flexible Diffuser

was varied in length and outlet diameter, as shown in Figure 16. The skirt material was nylon cloth covered with neoprene. In order to resist movement of the diffuser caused by turbulence, a steel ring was installed at the bottom of the diffuser. The ring allowed tying the diffuser to anchors located on the bottom of the lake.

Supporting Structure

As shown in Figure 17, the pump was held by a supporting frame having a length of 72 inches. This dimension allowed the propeller blade to operate at a depth of four feet below the surface. The frame was connected to the floating platform by means of a pair of gimbals allowing the pumping unit to remain stationary as the platform moved in the water (Figure 18). On top of the supporting frame the electric motor, pulley and shaft bearing were installed.

Platform

In order to locate and hold the pumping unit, an 8.0 ft. by 16.0 ft. wooden raft was built. The raft floated on 10 fifty-five gallon barrels. Total supporting capacity of the raft was 2,500 lbs and four anchors were placed off the four corners of the raft. The floor of the raft was installed in such a way that it could be easily removed to allow the pump to be raised when changes of variables were required. An 8.0 ft. A-frame with a winch was used to lift the pump (Figure 19).

Power Sources

Power supply and control consisted of: a) A raft mounted electric generator with 2.5 Kva capacity, b) A 1/2-Hp Dayton electric motor,

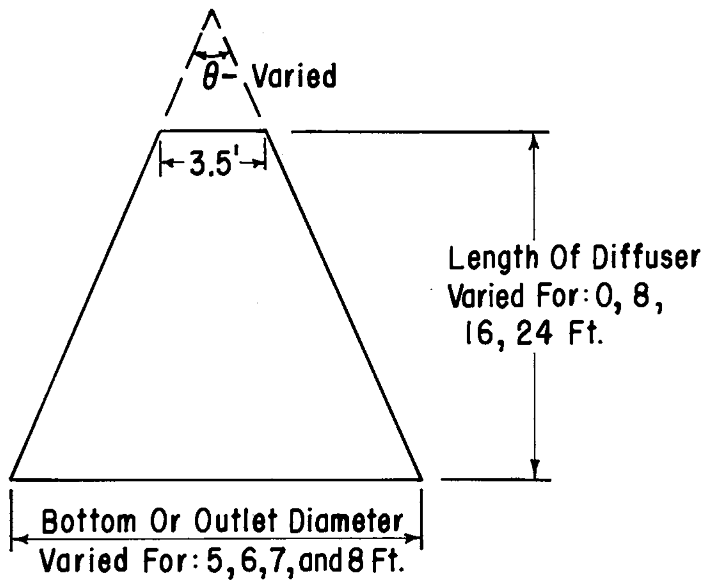

Figure 16. Arrangement of the Diffuser

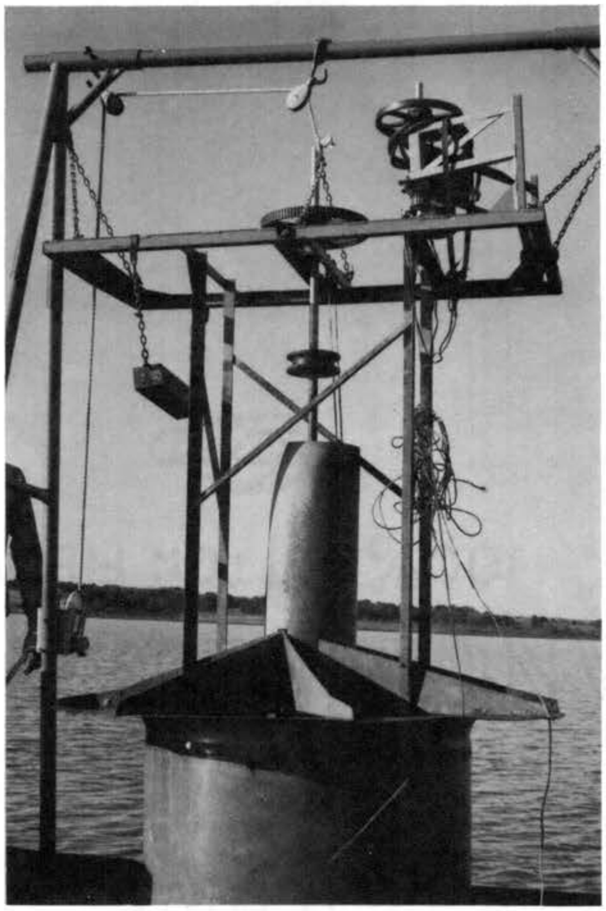

Figure 17. Supporting Structures

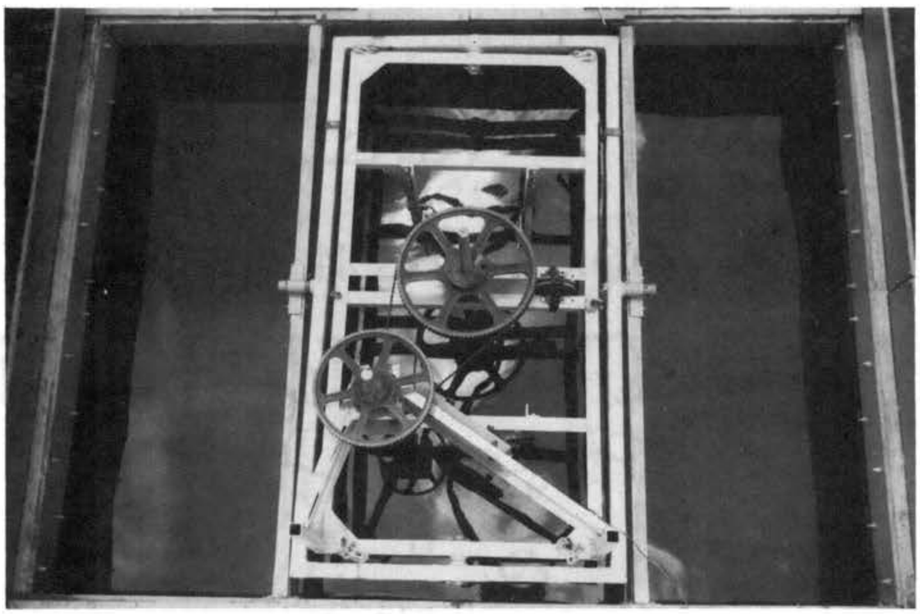

Figure 18. Supporting Frame with Gimbals

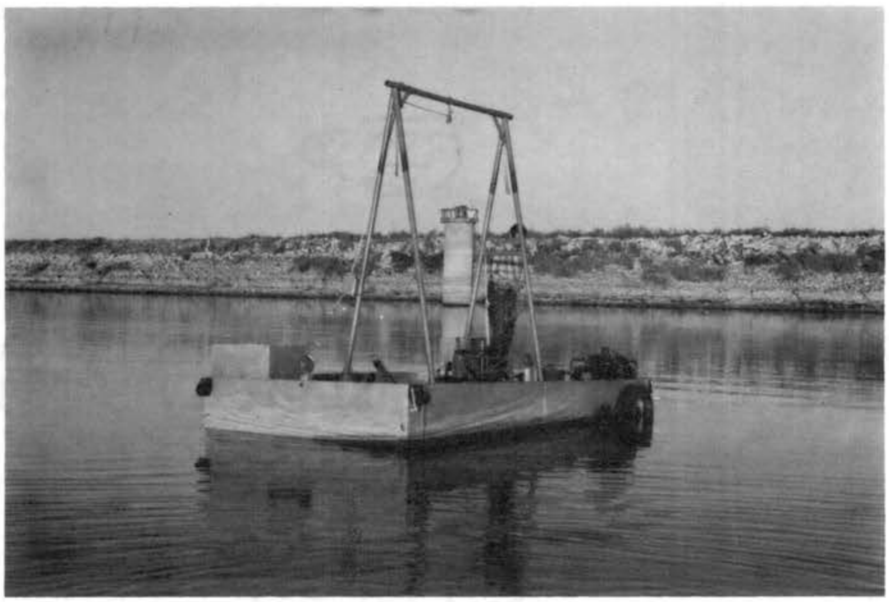

Figure 19. General View of the Raft

and c) A set of pulleys. As the propeller was belt driven, the set of four pulleys was installed as a speed reducer to obtain approximately 40, 56, 60 and 80 rpm. Positive drive (timing belts) was used to avoid slippage (Figure 20).

Tests were made using a prony brake to obtain the Hp input versus Hp output of the shaft (Figure 21).

Measuring Devices

For measuring velocity of the water through the flume, a laboratory "OTT" current meter with a propeller 50 mm in diameter and a 0.05 pitch was selected. Velocity was determined by using a revolution counter, calibrated for measuring ranges in m/sec from 0.05 to 3.0 (Figure 22). Propellers were factory calibrated so the equations as well as calibration curves were used to calculate the velocity of the water based on the number of revolutions of the propeller. In order to obtain the average velocity, a system proposed by Henderson (30) was followed. It consisted of dividing the conduit in six equal concentric areas, in which velocity was measured at the center of each area, and an average of the six was taken as the velocity for the total sectional area (Figure 23).

For measuring power, an ammeter, voltmeter, and small-scale wattmeter was assembled as shown in Figure 24.

Location of the Experiment

Lake Carl Blackwell, located 10 miles west of Stillwater, Oklahoma, was selected as the location for the experiments. A map of contours was obtained and depths were checked by a sounding device. A

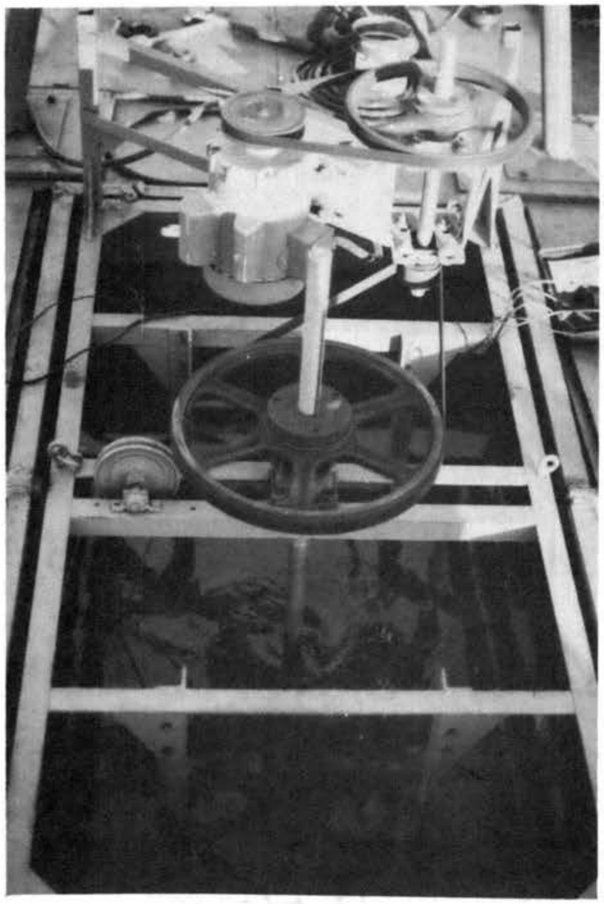

Figure 20. Assembly of Pulleys and Motor

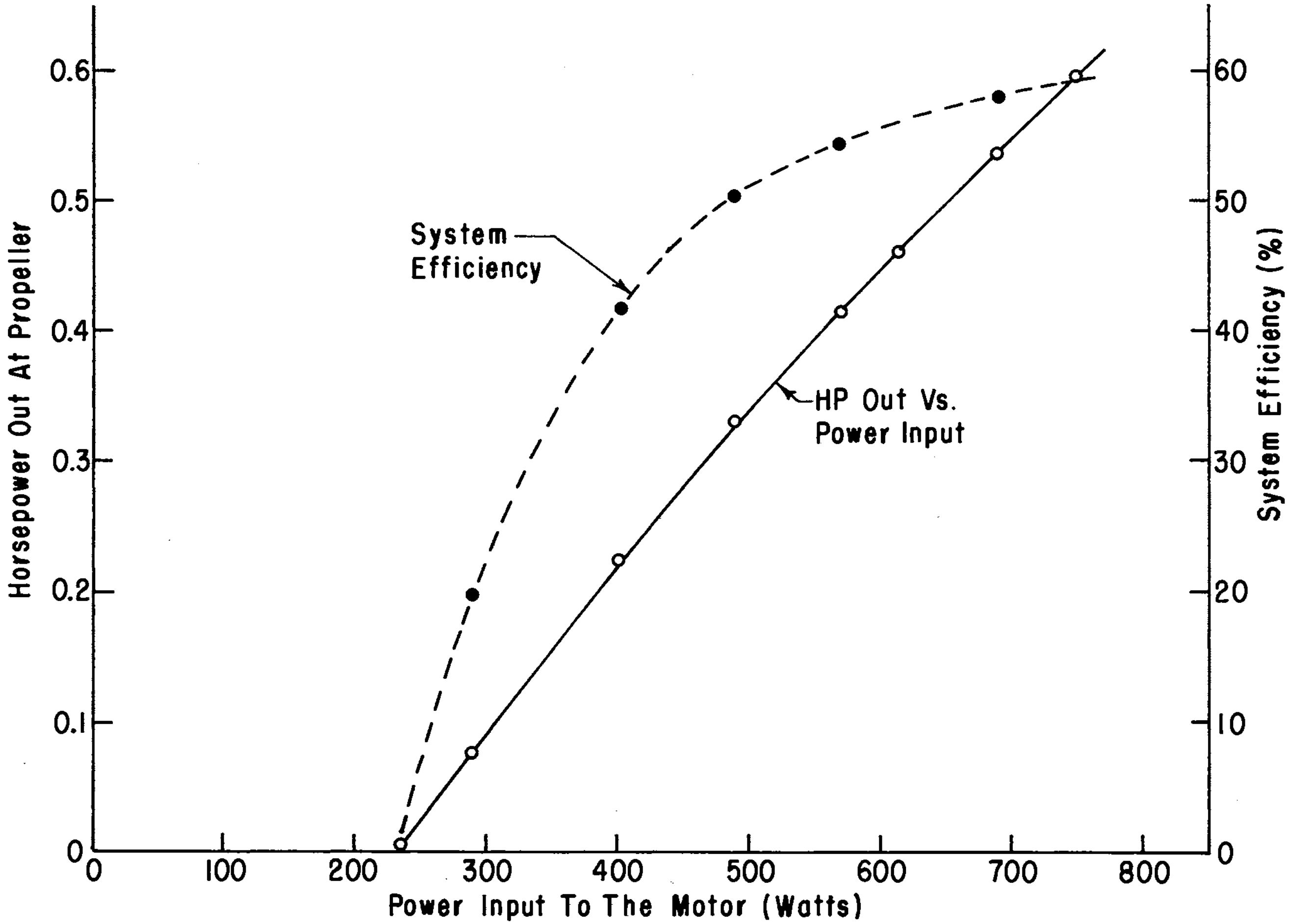

Figure 21. Results of Prony Brake Tests

Figure 22. Measurement of Velocity

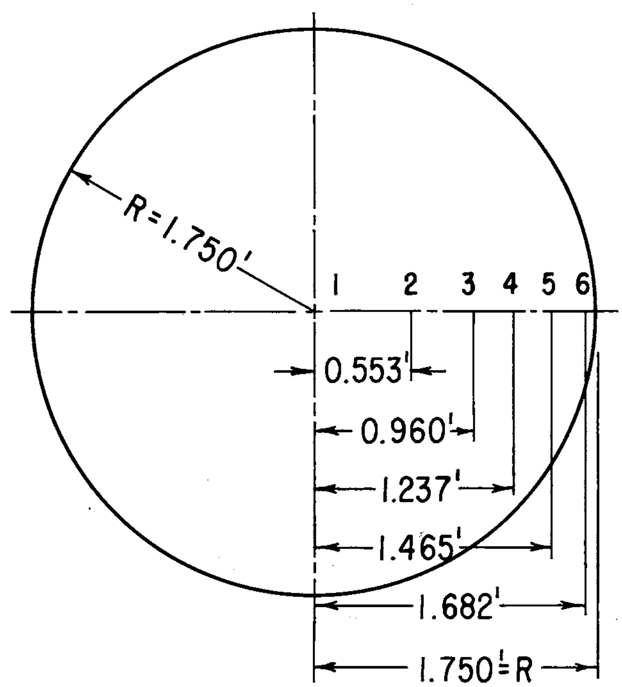

Figure 23. Points where Velocity was Measured at Pump Section

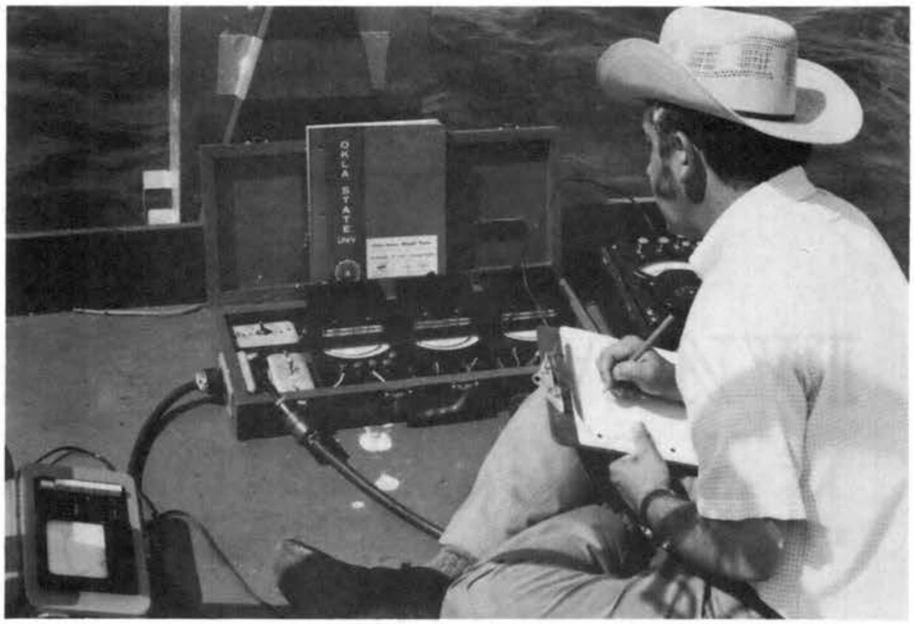

Figure 24. Power Measuring Instruments

location having a depth of about 31 ft. was selected and marked with a bouy. The reasons for selecting Lake Blackwell were: a) Owned by Oklahoma State University, b) Located near the mentioned Institution and c) Provided easy accessibility for the equipment and personnel.

CHAPTER IV

METHODS AND PROCEDURE

Design of the Experiment

The design of the experiment consisted of setting treatment levels to respectively determine the relationships of Flow (Q) and Horsepower (Ps) to: a) propeller rpm, b) diffuser outlet diameter and c) diffuser length.

The experimental design was a complete factorial. The dependent variables were the velocity of water at the throat, and the power required. The independent variables were: three lengths of diffuser, four outlet diameters of diffuser, and four pulleys that represented four propeller shaft velocities.

It would have been desirable to completely randomize the order of the variables in each of the experiments. However, this was not practical because the diffuser had to be cut when varying the length conditions. The following randomization procedure was used: a) the order of the outlet diameter was randomized, b) the order of the pulley used was randomized. Concerning the length value, experiments were run with the 24 ft. length first, then the 16.0 ft. and finally the 8.0 ft. The tests were conducted according to the experimental design shown in Table I.

Observations of velocity, rpm and watts were made for every condition in which length of the skirt, outlet diameter of the diffuser and pulley size were varied. Inlet diameter was kept constant at a value of 3.5 ft.

Measurements on the Experimental Plan

The following measurements were recorded for each test:

1. Revolutions per second of the flow meter were recorded to the nearest 0.1 rps. One value was recorded for each position of the flow meter. Then, using the calibration curve, the values were converted to corresponding velocities in ft./sec. An average of the six readings for six equal areas, was taken as the velocity for the test.

2. Power input in watts was recorded to the nearest 1.0 watt. Using the calibration curve from the prony brake test (Figure 21), values for horsepower output of the shaft were determined.

3. Using the belt from the jackshaft to the pump, linear velocity was measured to the nearest 1.0 ft./min. Based on the laboratory tests, a conversion factor was determined to obtain propeller rpm. The propeller shaft rpm was equal to the belt speed in feet per minute multiplied by 0.052.

TABLE I

THE EXPERIMENTAL DESIGN

Test No.	Run No.	Inlet Diameter Di (Ft)	Diffuser Length L (Ft)	Outlet Diameter Do (Ft)	Pulley No.
1	36	3.5	8.0	5.0	20
2	33				28
3	34				30
4	35				40
5	47			6.0	20
6	46				28
7	45				30
8	48				40
9	41			7.0	20
10	43				28
11	44				30
12	42				40
13	37			8.0	20
14	39				28
15	40				30
16	38				40
17	32		16.0	5.0	20
18	29				28
19	30				30
20	31				40
21	17			6.0	20
22	19				28
23	20				30
24	18				40
25	26	3.5	16.0	7.0	20
26	28				28
27	27				30
28	28				40

TABLE I (Continued)

Test No.	Run No.	Inlet Diameter Di (Ft)	Diffuser Length L (Ft)	Outlet Diameter Do (Ft)	Pulley No.
29	23			8.0	20
30	24				28
31	21				30
32	22				40
33	16		24.0	5.0	20
34	13				28
35	14				30
36	15				40
37	7			6.0	20
38	6				28
39	8				30
40	5				40
41	11			7.0	20
42	12				28
43	10				30
44	9				40
45	2			8.0	20
46	3				28
47	1				30
48	4				40
49	52	No Skirt			20
50	49				28
51	50				20
52	51				40

CHAPTER V

PRESENTATION AND ANALYSIS OF DATA

The raw data from the experimental program were the values measured and calculated as velocity of flow at the throat (V_1), Propeller shaft speed (RPM), and the horsepower input (Hp) for the conditions established in the experimental plan.

The data in terms of independent and dependent variables were analyzed by partitioning sum of squares, conducting F tests on mean squares, developing prediction equations using multiple regression, and by plotting graphs involving selected variables.

The analysis used was a Statistical Analysis System, SAS, for reduction of sum of squares and also for the regression using the 360-65 digital computer.

Velocity Variations

Velocities at the throat ranged from 1.295 fps that corresponds to a flow of 12.45 cfs or 5,590 gpm to 2.48 fps that corresponds to a flow of 23.9 cfs or 10,700 gpm.

The values correspond to the following conditions:

	V 1.295 fps	V 2.48 fps
Length of diffuser, Ft	24.0	16.0
Diffuser outlet diameter, Ft	5.0	8.0
Propeller shaft velocity, RPM	41.5	76.3
Propeller shaft Horsepower, Hp	0.045	0.50
Motor input, Watts	265.0	650.0

Measured velocities of each of the tests are presented in Table XII of Appendix A.

The flow (Q) values for corresponding values of rpm of each of the tests are presented in Figures 25, 26 and 27. In each test a least squares best fit to a polynomial equation, expressing flow through the pump, as a function of propeller shaft angular speed (rpm), was obtained. The general form of the polynomial equation is:

$$Q_N = \beta_o + \beta_1 N \tag{5-1}$$

where:

Q_N = Flow related to N rpm, cfs.

N = Propeller shaft angular speed, rpm.

β_o = Regression coefficient.

β_1 = Regression coefficient.

Data for the entire tests were used to determine a least squares best fit curve for test results presented in Figures 25 through 27. Results of each of the general tests are given in Table II.

Power Variations

The values for power in Hp. for corresponding values of propeller shaft angular speed (rpm) of each of the tests are presented in

TABLE II

REGRESSION COEFFICIENTS OF FLOW (Q) VS. PROPELLER SHAFT VELOCITY (RPM) POLYNOMIAL EQUATIONS* OF VELOCITY TESTS

Diffuser Length (Ft)	Outlet Diameter (Ft)	β_o	β_1	Range Applicable (RPM)	R	Standard Deviation	Observations
0.0	3.0	3.863668	0.184093	40 - 80	0.996	0.2929	No Skirt Used
8.0	5.0	1.797317	0.271553	40 - 80	0.999	0.1371	
8.0	6.0	1.362823	0.282133	40 - 80	0.999	0.0918	
8.0	7.0	0.927307	0.293677	40 - 80	0.999	0.1415	
8.0	8.0	0.984603	0.296907	40 - 80	0.999	0.0428	
16.0	5.0	2.142501	0.263142	40 - 80	0.999	0.1812	
16.0	6.0	0.767486	0.297988	40 - 80	0.999	0.0272	
16.0	7.0	0.685623	0.302282	40 - 80	0.999	0.0585	
16.0	8.0	0.868606	0.301505	40 - 80	0.999	0.0891	
24.0	5.0	1.221679	0.271648	40 - 80	0.999	0.0789	
24.0	6.0	1.295028	0.274091	40 - 80	0.999	0.1105	
24.0	7.0	1.385833	0.275306	40 - 80	0.999	0.1030	
24.0	8.0	1.179763	0.286611	40 - 80	0.999	0.1159	

* $Q = \beta_o + \beta_1$ (RPM)

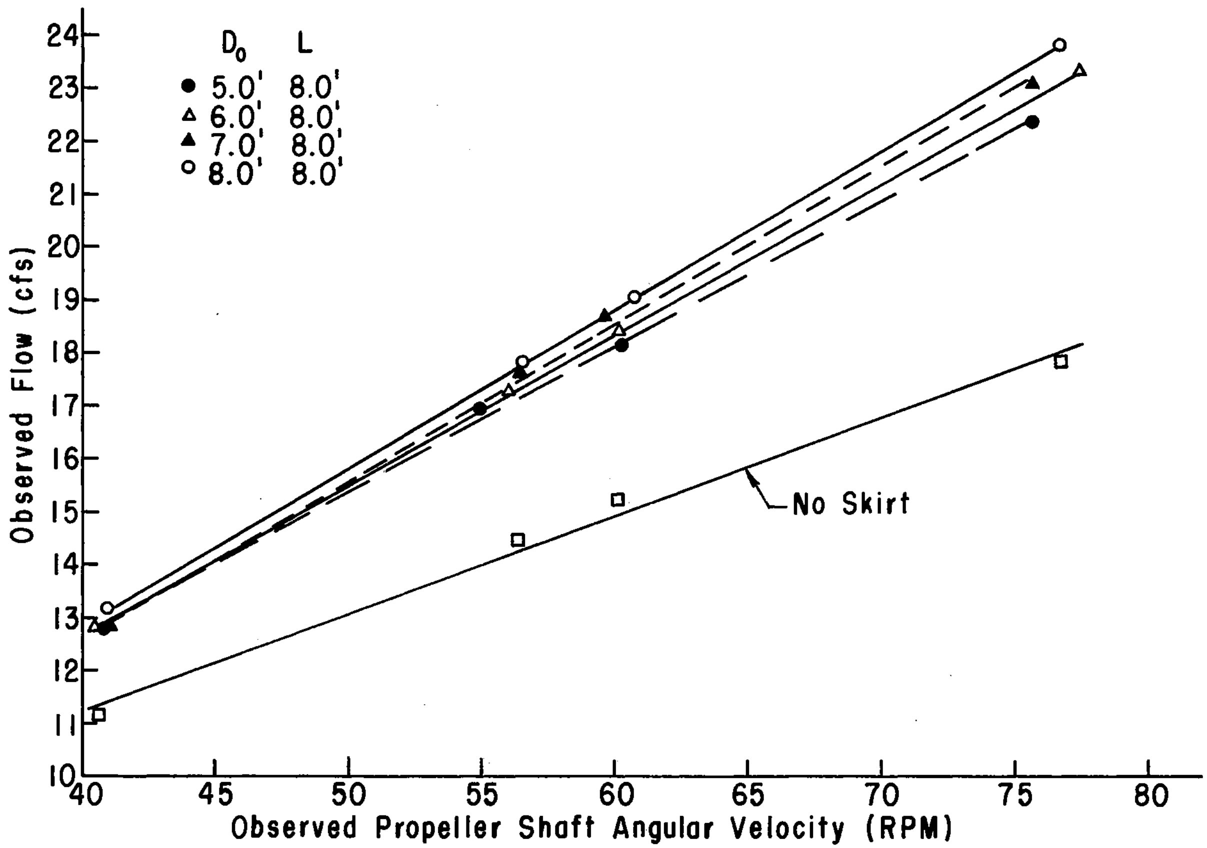

Figure 25. Variations in Flow for Different Propeller Velocities with 8.0 Foot Long Diffuser and Four Outlet Diameters

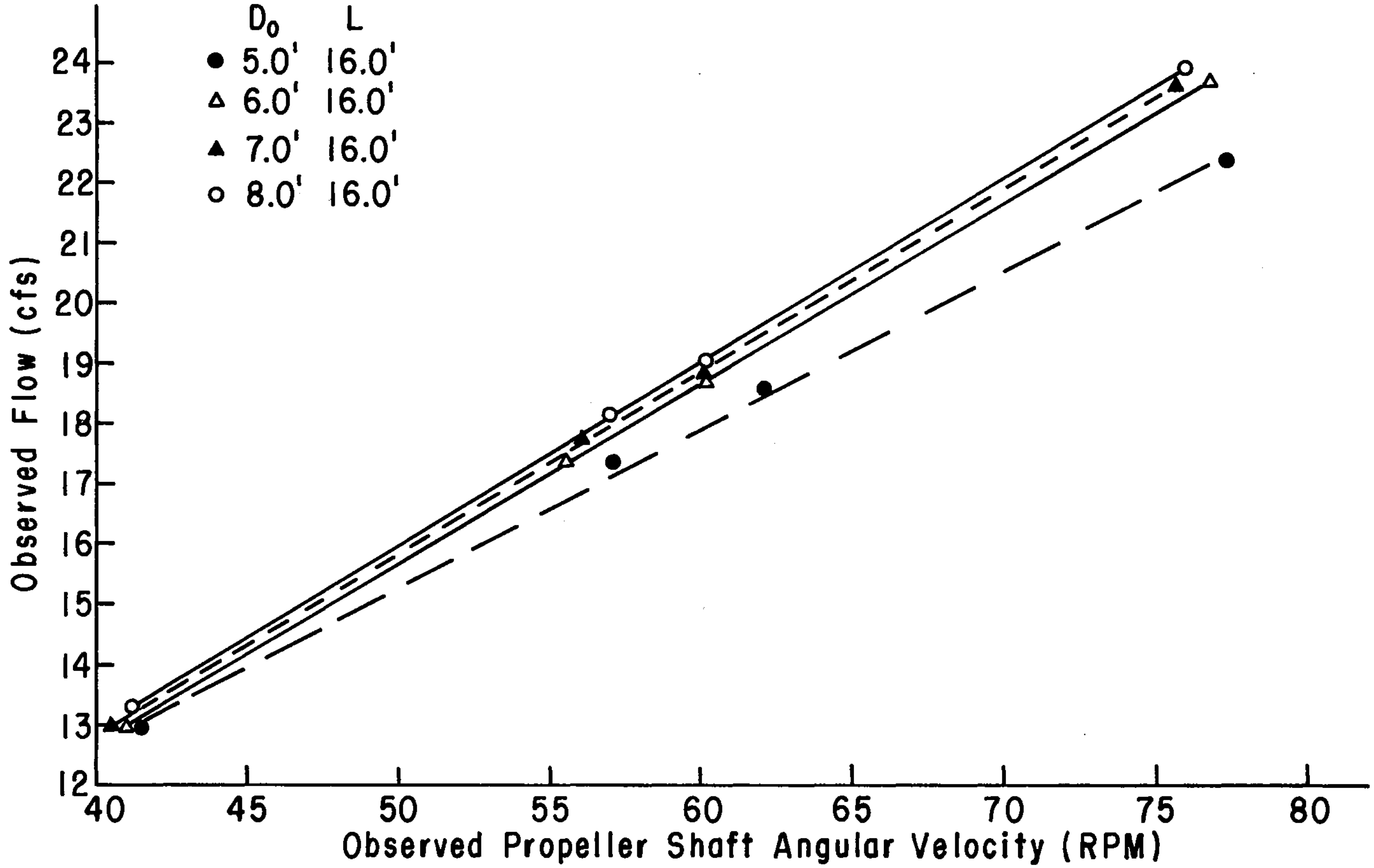

Figure 26. Variations in Flow for Different Propeller Velocities with 16.0 Foot Long Diffuser and Four Outlet Diameters

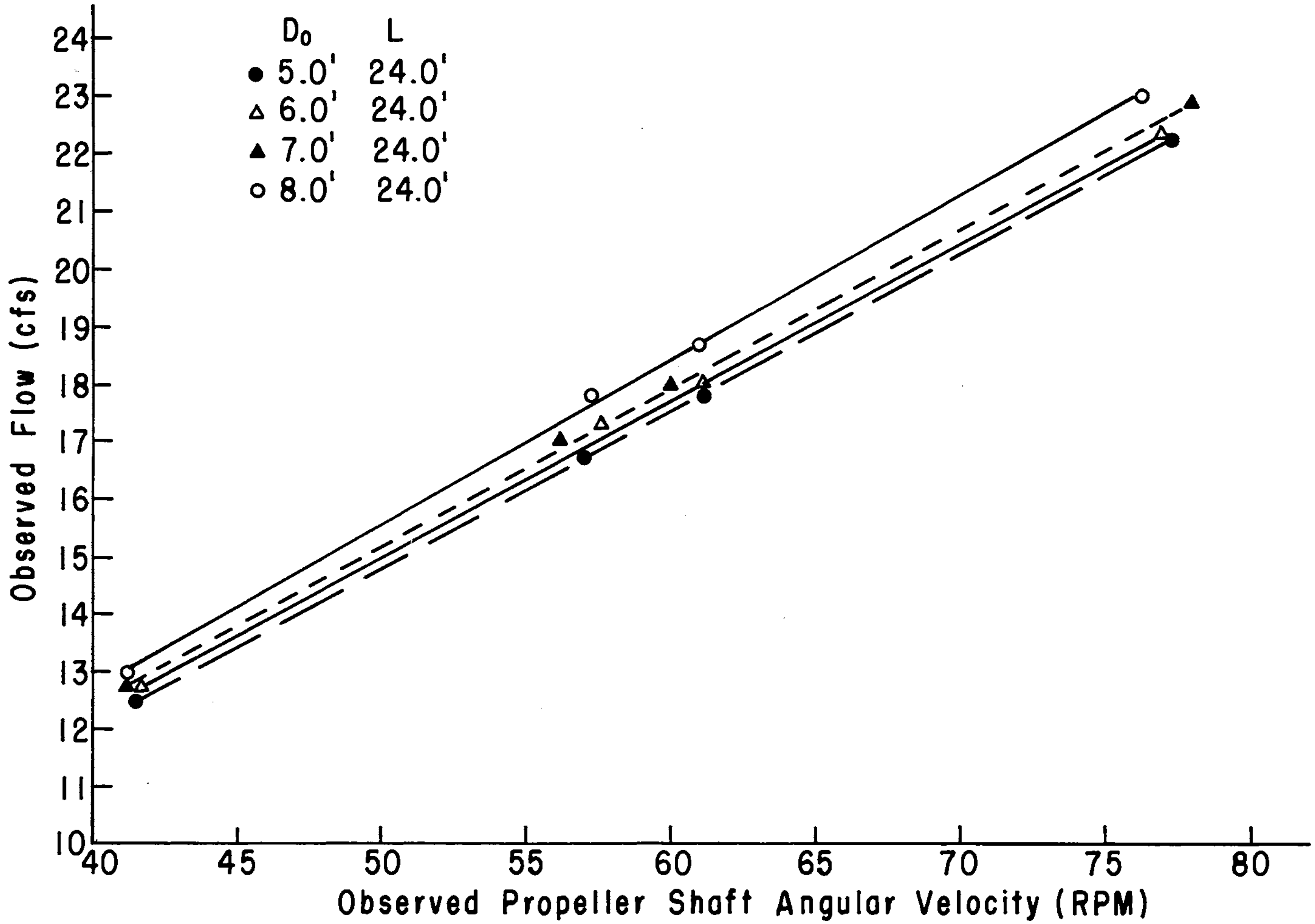

Figure 27. Variations in Flow for Different Propeller Velocities with 24.0 Foot Long Diffuser and Four Outlet Diameters

Figures 28 through 31. In each test a least squares best fit for a polynomial equation, expressing horsepower for a certain flow through the pump as a function of propeller shaft angular speed was obtained. The general form of the polynomial equation is as follows:

$$P_s = \beta_o + \beta_1 \times N^3 \qquad (5\text{-}2)$$

Where:

P_s = Propeller shaft power, Hp.

N = Propeller shaft angular speed, rpm.

β_o = Regression coefficient.

β_1 = Regression coefficient

Data for the entire tests were used to determine a least squares best fit curve for tests presented in Figures 28, 29, 30 and 31. Results of each of the general tests are given in Table III. The standard deviations and the applicable range for each equation is presented in Tables IV and V.

In all the flow versus propeller velocity and power versus propeller velocity curve fitting efforts, the degree of the equation selected was based on: a) the standard deviation, b) the correlation coefficient R and c) the relative degree of goodness of fit of the equation.

Analyses of variance were performed using individual observations. The sum of squares partitioning and F tests are presented in Tables IV and V. The information in these tables was obtained using the SAS program for factorial designs. The total variation associated with each dependent factor was partitioned among seven sources. These seven sources consisted of three main effects and four interactions.

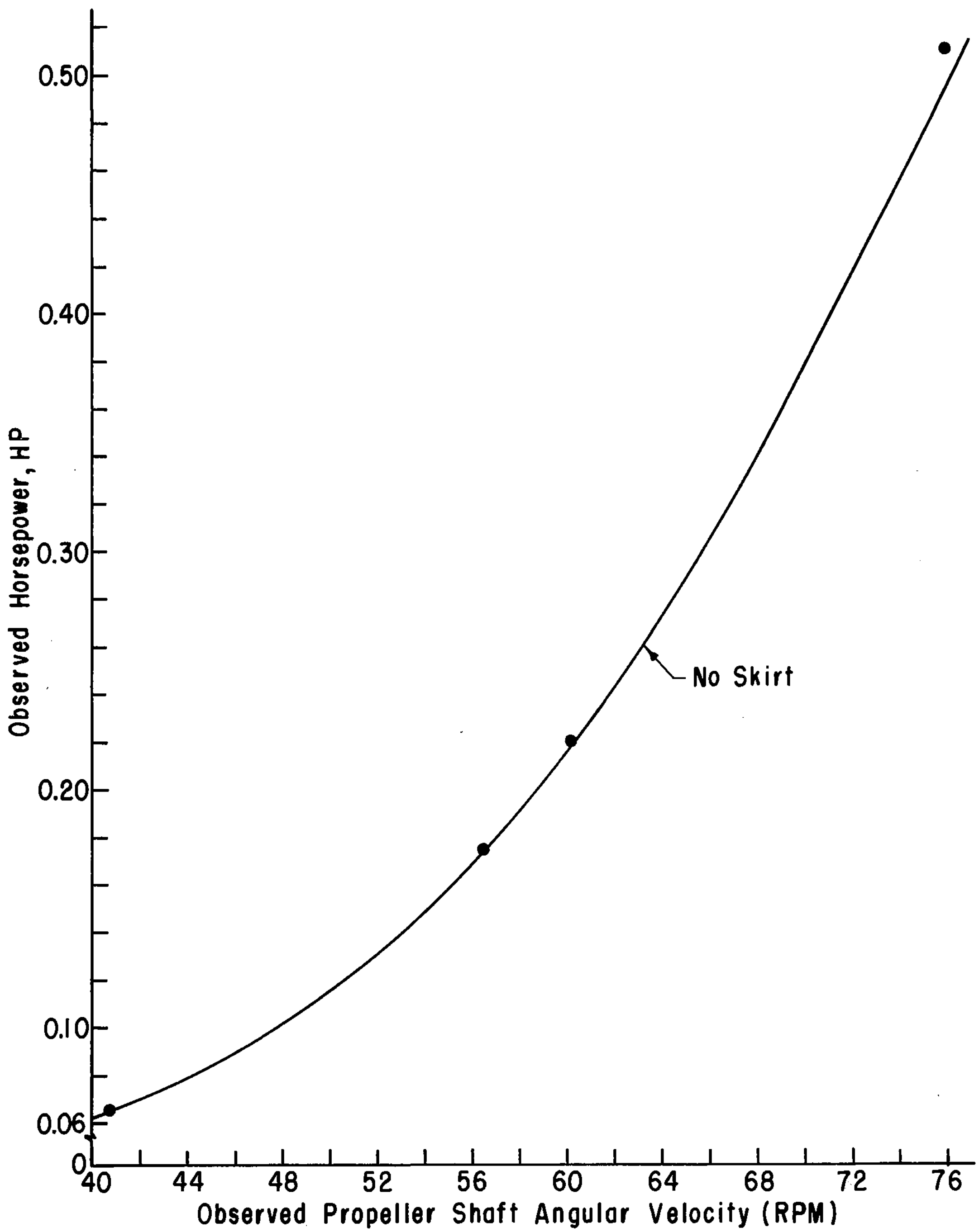

Figure 28. Variations in Power for Different Propeller Velocities with no Diffuser

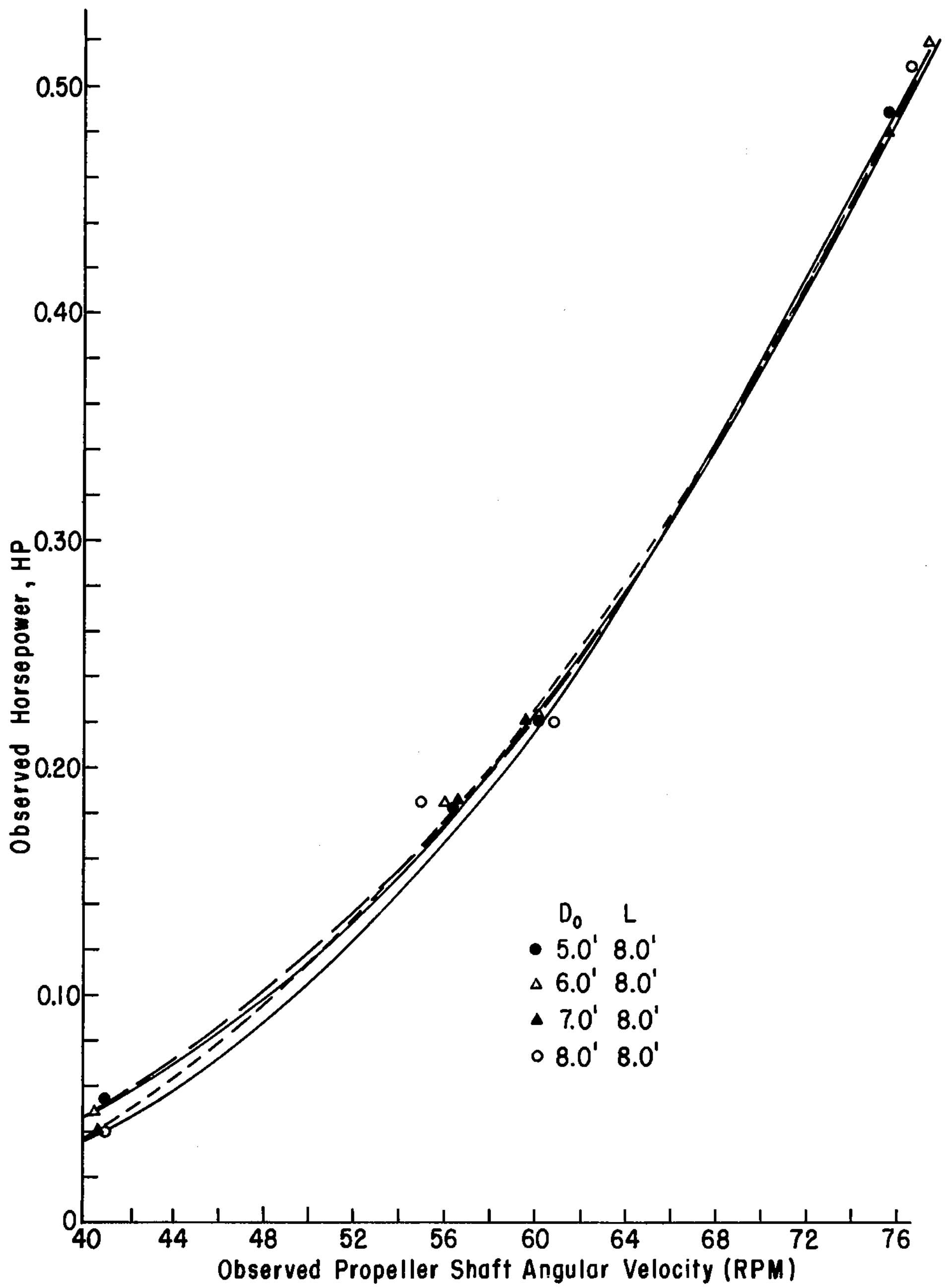

Figure 29. Variations in Power for Different Propeller Velocities with 8.0 Foot Diffuser with L = 8 Ft. and Four Outlet Diameters

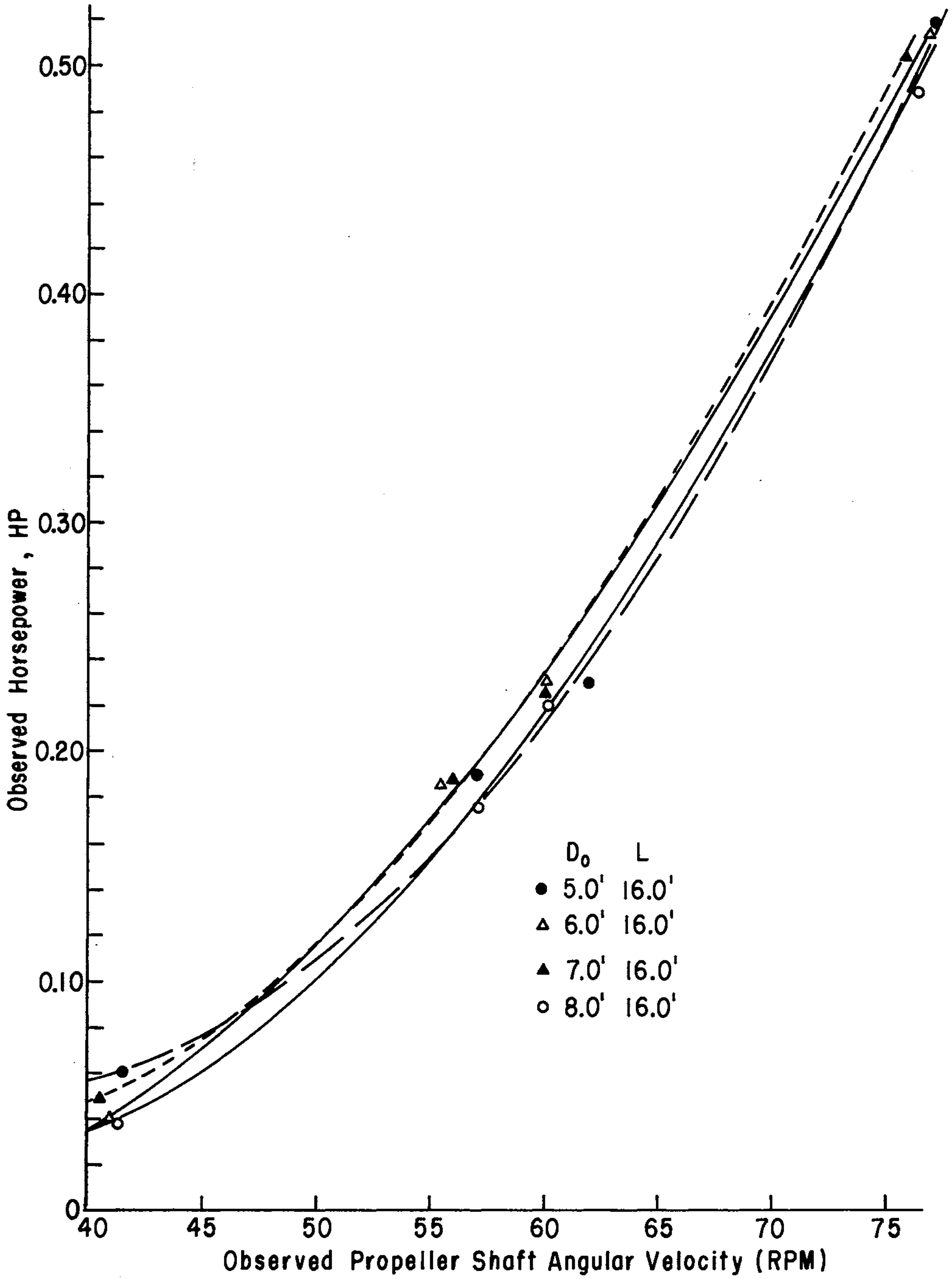

Figure 30. Variations in Power for Different Propeller Velocities with 16.0 Foot Diffuser with L = 16 Ft. and Four Outlet Diameters

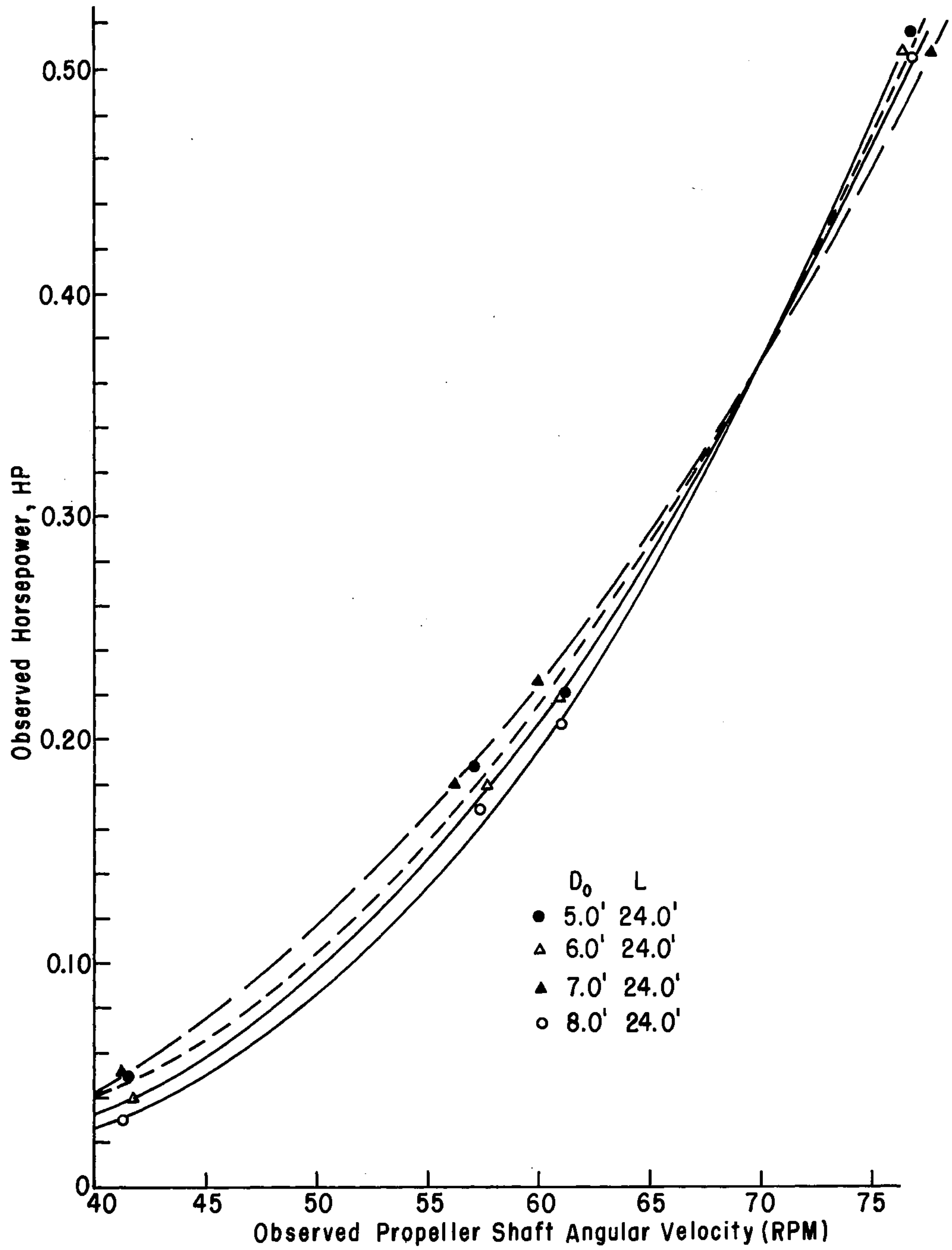

Figure 31. Variations in Power for Different Propeller Velocities with 24.0 Foot Diffuser, with L = 24 Ft. and Four Outlet Diameters

TABLE III

REGRESSION COEFFICIENTS OF POWER (Ps) VS. PROPELLER SHAFT VELOCITY (RPM) POLYNOMIAL EQUATIONS* OF POWER TESTS

Diffuser Length (Ft)	Outlet Diameter (Ft)	β_o	β_1	Range Applicable (RPM)	R	Standard Deviation	Observations
0.0	3.5	-0.0266030	0.0000012	40 - 80	0.998	0.0130	No Diffuser Used
8.0	5.0	-0.0240945	0.0000012	40 - 80	0.98	0.0130	
8.0	6.0	-0.0203615	0.0000012	40 - 80	0.999	0.0072	
8.0	7.0	-0.0353087	0.0000012	40 - 80	0.999	0.0053	
8.0	8.0	-0.0445641	0.0000012	40 - 80	0.999	0.0091	
16.0	5.0	-0.0321627	0.0000012	40 - 80	0.997	0.0152	
16.0	6.0	-0.0356281	0.0000012	40 - 80	0.999	0.0085	
16.0	7.0	-0.0374107	0.0000012	40 - 80	0.999	0.0085	
16.0	8.0	-0.0505057	0.0000012	40 - 80	0.999	0.0033	
24.0	5.0	-0.0442061	0.0000012	40 - 80	0.998	0.0140	
24.0	6.0	-0.0538356	0.0000012	40 - 80	0.999	0.0059	
24.0	7.0	-0.0224595	0.0000012	40 - 80	0.999	0.0076	
24.0	8.0	-0.0709364	0.0000013	40 - 80	0.998	0.0140	

$*Ps = \beta_o + \beta_1 (RPM)^3$

TABLE IV

ANALYSIS OF VARIANCE FOR VARIABLE Q, AND FACTORS LENGTH, L, DIAMETER, Do, AND PULLEY SIZE

Source	df	SS	MS	F
L	2	2.453882	1.226941	29.5*
DD	3	4.005404	1.335135	32.15*
L*DC	6	0.077096	0.012849	0.309
PULL	3	624.845792	208.281931	5015.9*
L*PULL	6	0.302377	0.050396	1.21
DD*PULL	9	0.896502	0.099611	2.39
L*DD*PULL	18	0.747427	0.041524	
CORRECTED TOTAL	47	633.328479	13.475074	

*Significant at the 0.5 percent level of significance

TABLE V

ANALYSIS OF VARIANCE FOR VARIABLE, Ps, AND FACTORS LENGTH, L, DIAMETER, Do, AND PULLEY SIZE

Source	df	SS	MS	F
L	2	0.00024679	0.000123396	2.84*
DD	3	0.00070156	0.000233854	5.38*
L*DD	6	0.00075488	0.000125813	2.89*
PULL	3	1.35272656	0.450908854	10388.4*
L*PULL	6	0.00032038	0.000053396	1.23
DD*PULL	9	0.00047952	0.000053280	1.22
L*DD*PULL	18	0.00078129	0.000043405	
CORRECTED TOTAL	47	1.35601098	0.028851297	

*Significant at the 0.10 percent level of significance

Performed F tests were made in a standard manner for all levels.

In the analysis of variance for the variable Flow (Q), the effect of three-factor interaction was 0.748 and counted for 0.118% of the total variation.

In the analysis of variance for the variable Power (Ps), the effect of three-factor interaction was 0.00078 and counted for 0.05% of the total variation.

Three-factor interactions were assumed to not exist; thus it was possible to use the L x Do x N mean squares as error to test the main effects and two-factors interaction. For Flow (Q), the effect of propeller shaft angular speed accounted for 98.66% of the total variation and for the variable Power (Ps), propeller shaft angular speed accounted for 97.7% of the total variation.

Prediction Equations for Flow Q and Power Ps

Equation 5-3, for flow rate, was obtained by multiple regression analysis of all data obtained in this study.

$$Q = -2.07001404 + 0.27627960\ N + 0.54165381\ Do + 0.00920268\ L \tag{5-3}$$

$$R^2 = 0.9621$$

where:

Q = Flow through the pump, cfs.

N = Propeller shaft angular speed, rpm.

Do = Outlet diameter of the diffuser, ft.

L = Length of the diffuser, ft.

R^2 = Squared value of the correlation coefficient.

Similarly, an equation was obtained for the Power:

$$Ps = -0.02433906 + 0.0000012\ N^3 - 0.00112998\ Do - 0.00047625\ L \qquad (5\text{-}4)$$

$$R^2 = 0.9958$$

where:

Ps = Propeller shaft power, Hp.

Flow calculated from equation 5-3 is plotted against Flow observed in Figure 32. Similarly, Power calculated from equation 5-4 is plotted against Power observed in Figure 33.

Head Losses Through The Device

Recalling Bernoulli's equation (2-3), head loss through the diffuser depend on: a) Type of entrance, b) Effect of the enlargement and c) The exit of the diffuser.

Loss Due to Entrance, H_1

Hamilton (22) studied the entrance of a fluid into pipes when a well shaped bell-mouth entrance was carefully built. He established a curve showing the decrease of entrance loss with the increase of intake rounding. Entrance loss was expressed in terms of velocity head as a function of the ratio of radius of entrance rounding to pipe diameter (R/D). The entrance of the flume under study presents similar characteristics to those mentioned by Hamilton. The conditions for the present study are:

Diameter of entrance: 42 inches

Radius of curvatures: 2-3/4 inches

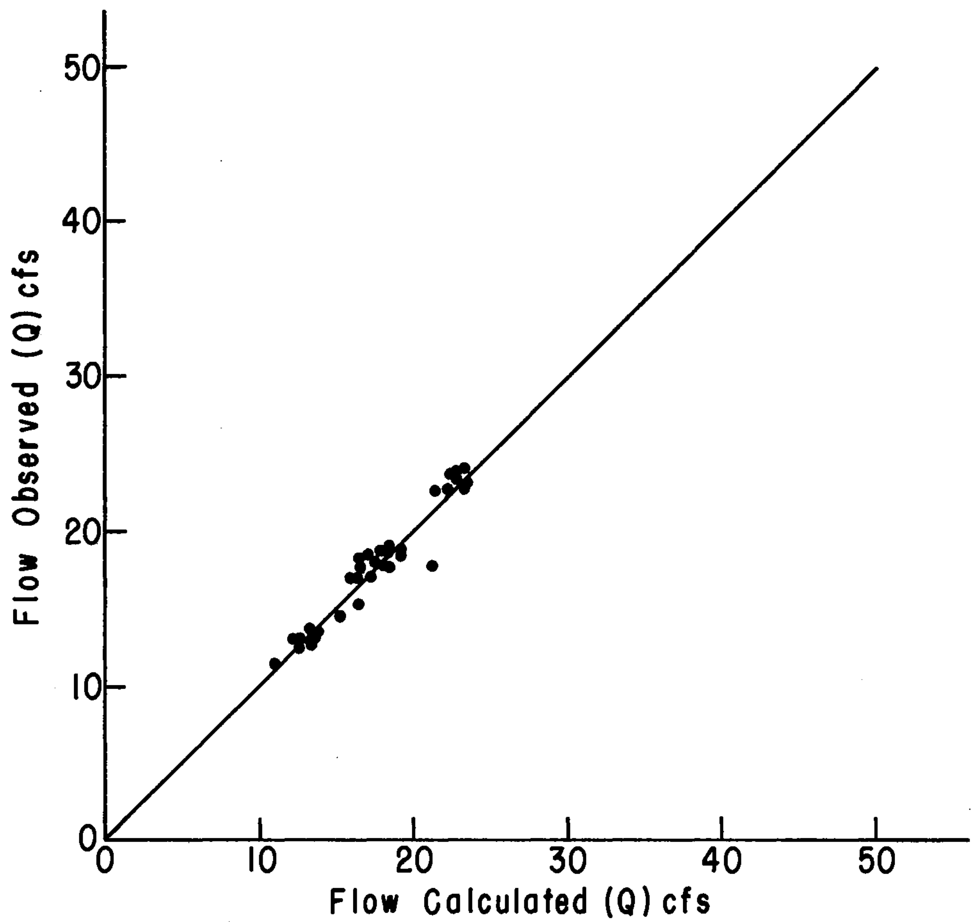

Figure 32. Relationship Between Observed and Calculated Flows

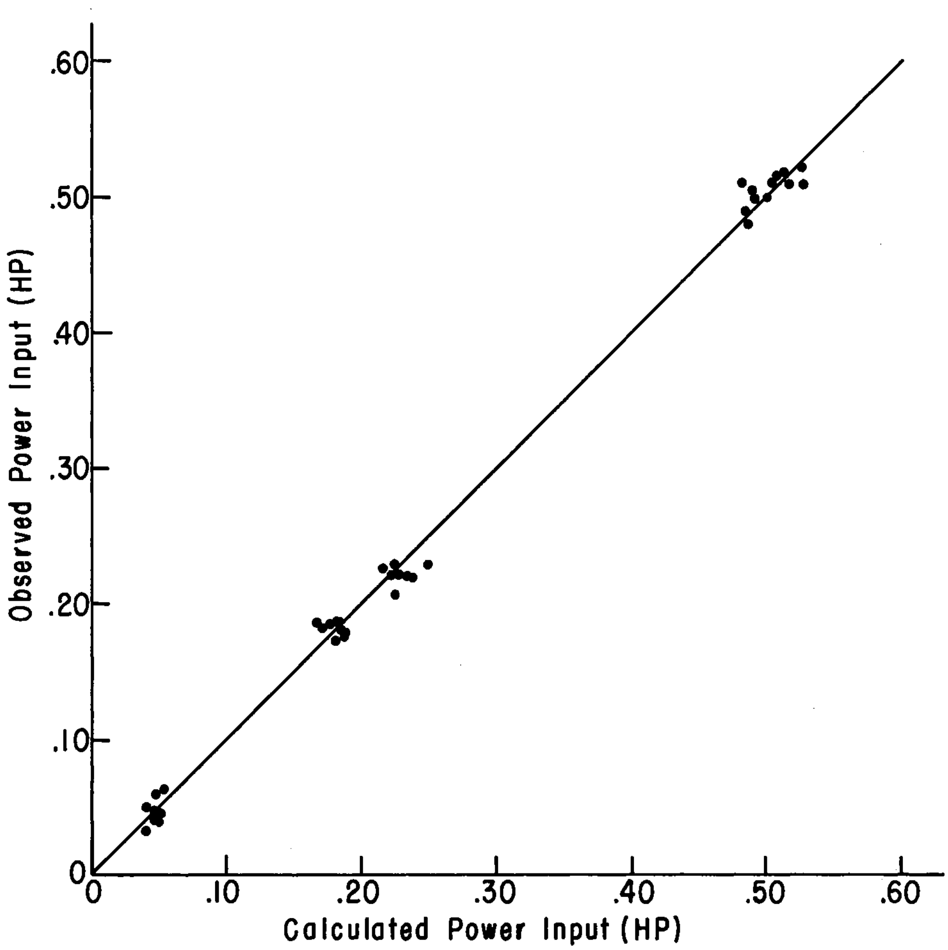

Figure 33. Relationship Between Observed and Calculated Power

As R/D corresponds to a value of 0.065, the reference value for

$K_1 = 0.10$ and $H_1 = 0.10\ (V_2)^2/2g$ (5-5)

Loss Due to Enlargement, H_2

The loss of head in the expanding jet is due to friction and fluid turbulence and its coefficient depends upon the cone angle and the area ratio. From Vennard (24), values of K_2 were established for several angles (Table VI). They can be used in equation 5-6 to determine the head loss.

$$H_2 = K_2(V_2 - V_3)^2/2g \qquad (5\text{-}6)$$

where:

H_2 = Head loss due to enlargement, ft.

K_2 = Coefficient for sudden expansions, listed in Table VI.

V_2 = Velocity before the widening commences, fps, listed in Table VIII.

V_3 = Velocity after the widening has ceased, fps, (calculated).

Calculated values of H_2 are presented in Table IX of Appendix A.

Loss Due to Exit, H_3

From Chapter II, it was assumed that the coefficient K_3 equals 1.0. Thus, the corresponding value for the loss due to exit are equal to the velocity head at the outlet; then

$$H_3 = (1.0)\ \frac{(V_3)^2}{2g} \qquad (5\text{-}7)$$

TABLE VI

VALUES OF COEFFICIENT K_2 BASED ON ANGLES OF DIFFUSERS FOR LENGTHS AND DIAMETERS USED IN THE EXPERIMENT

Diffuser Length (Ft.)	Diffuser Outlet Diameter (Ft.)	Angle $\emptyset$ Degrees	Coefficient K_2
8.0	5.0	10.63	0.25
8.0	6.0	17.35	0.36
8.0	7.0	23.63	0.47
8.0	8.0	29.36	0.60
16.0	5.0	5.36	0.18
16.0	6.0	8.88	0.22
16.0	7.0	12.33	0.28
16.0	8.0	15.70	0.33
24.0	5.0	3.58	0.16
24.0	6.0	5.95	0.18
24.0	7.0	8.30	0.22
24.0	8.0	10.61	0.25

Total Head Loss, H_T

The total head loss is the summation of H_1, H_2 and H_3. This value is the total dynamic head, TDH, caused by the pump and diffuser. This total dynamic head affects the power requirements of the pump, and can be used to determine the efficiency of the pump for measured values of flow and power. Table VIII gives the values of each type of loss and the corresponding total head loss for each test. The maximum calculated value of total head loss was 0.05 foot.

Estimation of Pump Efficiency

Based on calculated values of head, a determination of pump efficiency can be made for measured values of flow and power. Recalling equation 2-15, Chapter II, the expression for efficiency is:

$$\text{Efficiency} = \frac{Q \times H_T}{3960 \times Hp} \qquad (5\text{-}8)$$

where:

Q = Flow through the pump, gpm.

H_T = Total dynamic head, ft.

H_p = Power input, Hp.

Efficiencies calculated for each of the test flows are shown in Table IX. The highest efficiency found was 45.8 percent.

Estimation of Specific Speed

Conventional axial flow pumps have typical specific speeds of 7,500 to 14,000. Very low specific speeds are most often encountered in connection with small centrifugal pumps. For economic reasons it is necessary to select the specific speed always as high as possible, because for a given capacity and head, the dimensions of the pump will be smaller the higher the specific speed (28). Table X shows calculated specific speed values for each test. The values varied from 72,000 to 125,000 and are plotted in Figure 34. They were about 1 order of magnitude greater than those of a typical axial flow pump. As shown in Figure 34, extrapolated values of efficiency versus specific speed give rather low values of efficiency for a propeller pump at high specific speeds. However, no references were found which listed values of specific speed as high as those prevailing in these tests.

Dimensionless Parameters

In order to assist in the design and development of larger models, laws of Engineering similitude were considered. Mathematical analysis through the Buckingham Pi-theorem was followed. The Buckingham Pi-theorem reduces the number of experiments by reducing the parameters to dimensionless terms (31) allowing the formation of general equations and the development of a prediction equation for the performance of one dimensionless parameter as a function of the other parameters of the system.

The physical system for predicting the functioning of the pump may be adequately described by the pertinent quantities described in Table VII. The Table shows variable number, symbol, description, units and dimensional symbol. The power variable Ps is the dependent variable and is the quantity to be obtained.

Following the so called Rayleigh method of solving dimensional systems (32) and using the three fundamental dimensions of Mass M,

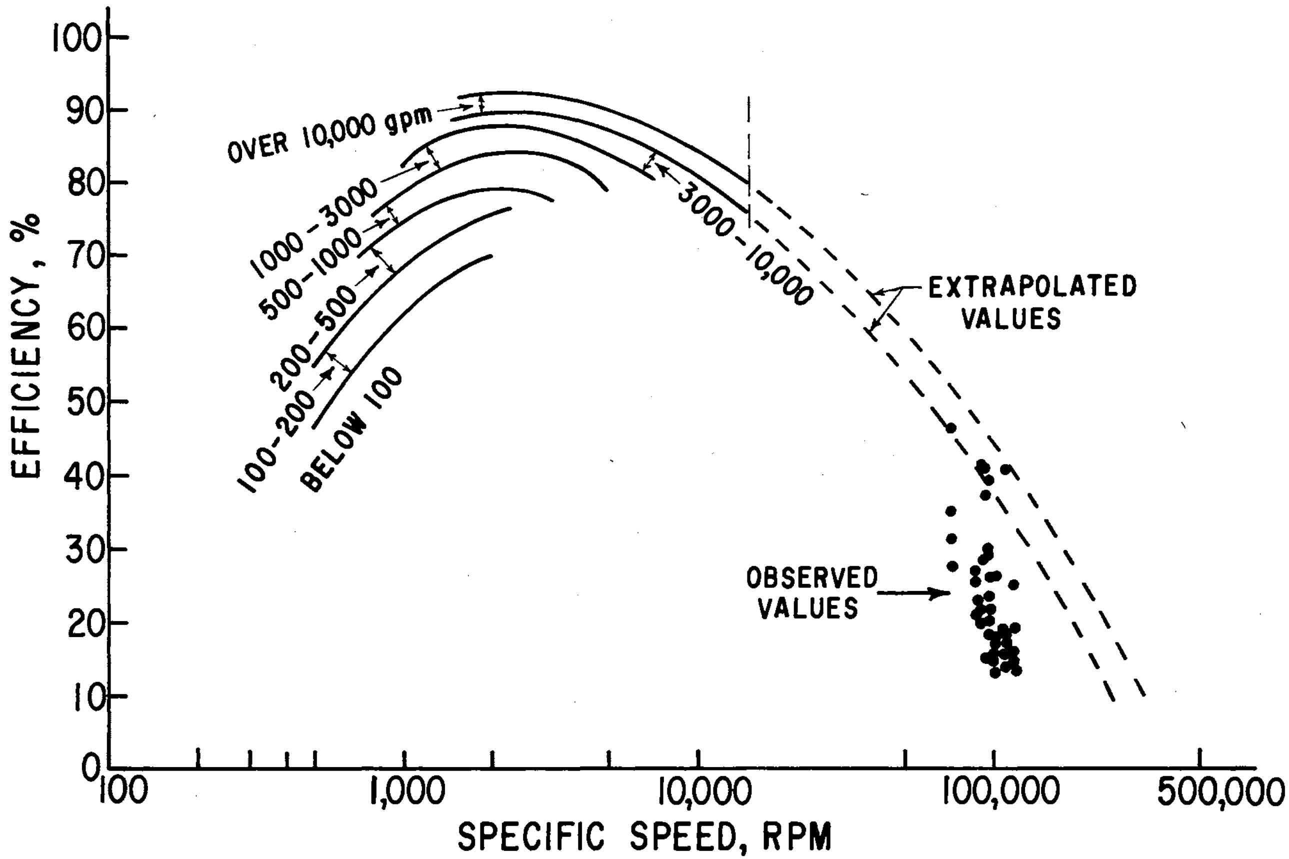

Figure 34. Calculated Values of Specific Speed of Pump

TABLE VII

LIST OF PERTINENT QUANTITIES

No.	Symbol	Description	Units	Dimension Symbol
1	μ	Fluid viscosity	Slugs/(Ft-sec)	$M\ L^{-1}\ T^{-1}$
2	ρ	Fluid density	$Slugs/Ft^3$	$M\ L^{-3}$
3	d	Pump diameter	Ft	L
4	D	Diffuser outlet diameter	Ft	L
5	L	Length of duffuser	Ft	L
6	ϵ	Diffuser roughness	Ft	L
7	P	Shaft horsepower	$Slug\text{-}Ft^2/Sec^3$	$M\ L^2\ T^{-3}$
8	Q	Fluid flow rate	Ft^3/Sec	$L^3\ T^{-1}$
9	g	Acceleration due to gravity	Ft/Sec^2	$L\ T^{-2}$
10	N	Propeller rotational speed	RPS	T^{-1}
11	H	Total head	Ft	L

Time T, and Length L, it is assumed that there exists a relationship among the variables such that:

$$f(\mu, p, d, H, P, Q, g, N, L, D, \epsilon) = 0 \tag{5-9}$$

Eleven quantities were required to describe the system. After using Huntley's addition, the rank of the dimensional matrix was found to be three. Therefore, a total of eleven minus three, or eight Pi-terms, was the absolute minimum number needed to describe the system.

The dimensionless parameters needed to investigate the power requirements are:

$$\pi_1 = \frac{d^4 P}{\rho Q^3} \tag{5-10}$$

$$\pi_2 = \frac{d^3 N}{Q} \tag{5-11}$$

$$\pi_3 = \frac{\mu d}{\rho Q} \tag{5-12}$$

$$\pi_4 = \frac{d^5 g}{Q^2} \tag{5-13}$$

$$\pi_5 = \frac{H}{d} \tag{5-14}$$

$$\pi_6 = \frac{D}{d} \tag{5-15}$$

$$\pi_7 = \frac{L}{d} \tag{5-16}$$

$$\pi_8 = \frac{\epsilon}{d} \tag{5-17}$$

Discussion of Pi-terms

a) $$\pi_1 = \frac{P d^4}{\rho Q^3} \tag{5-18}$$

or

$$\frac{P}{\rho d^5 N^3} \tag{5-19}$$

is called the Characteristic Power Number. Assuming that π_1 is constant, it follows that the corresponding power of a pump is:

$$P \approx N^3 \qquad \text{or} \qquad \frac{P_1}{P_2} = \frac{(N_1)^3}{(N_2)^3} \tag{5-20}$$

b) $$\pi_2 = \frac{Q}{d^3 N} \tag{5-21}$$

is called the Characteristic Discharge Number. It is an index of volumetric efficiency of the propeller. For a given pump with a constant diameter, the capacity varies directly with the speed of propeller rotation. If d is constant, then the discharge of the pump is directly proportional to the speed of its rotation.

$$Q \approx N \qquad \text{or} \qquad \frac{Q_1}{Q_2} = \frac{N_1}{N_2} \tag{5-22}$$

c) $$\pi_3 = \frac{\mu d}{\rho Q} \tag{5-23}$$

As γ is μ/ρ where γ = Kinematic viscosity of fluid, then

$$\pi_3 = \frac{\gamma d}{Q}, \qquad \text{or} \qquad \frac{\gamma}{d^2 N} \tag{5-24}$$

which is called the Characteristic Speed Number. It has been confirmed experimentally (33) that for turbulent flow through the propeller at high Reynolds numbers, the influence of the viscosity γ, is almost imperceptible.

d) $$\pi_4 = \frac{d^5 g}{Q^2} = \frac{d^4 gH}{Q^2} \tag{5-25}$$

that becomes

$$\pi_4 = \frac{gH d^4}{d^6 N^2} = \frac{gH}{d^2 N^2} \tag{5-26}$$

is called the Characteristic Head Number. If d is constant, then it follows that the corresponding head is:

$$H \approx N^2 \qquad \text{or} \qquad \frac{H_1}{H_2} = \frac{(N_1)^2}{(N_2)^2} \tag{5-27}$$

e) $\pi_5 = \frac{H}{d}$ Index of Head to diameter of the pump. (5-28)

f) $\pi_6 = \frac{D}{d}$ Ratio of diffuser outlet diameter to diameter of the pump. (5-29)

g) $\pi_7 = \frac{L}{D}$ Index of length of the diffuser to diameter of the propeller. This parameter fixes the influence of the length of the diffuser. (5-30)

h) $\pi_8 = \frac{\varepsilon}{d}$ Index of the roughness of the diffuser to the diameter of the pump; this relationship was considered constant. (5-31)

Similarity Conditions for Flow

When considering two geometrically similar pumps with propeller diameters d_1 and d_2 the following relationships can be obtained for two similar flows of liquids that have densities ρ_1 and ρ_2 and viscosities γ_1 and γ_2.

a) Conditions for discharge:

From equation 5-21 we have

$$\frac{Q_1}{Q_2} = \frac{d_1^3\ N_1}{d_2^3\ N_2} \tag{5-32}$$

b) Conditions for total head:

From equation 5-26 we have:

$$\frac{H_1}{H_2} = \frac{d_1^2\ N_1^2}{d_2^2\ N_2^2} \tag{5-33}$$

c) Conditions for Power:

From equation 5-19 we have:

$$\frac{P_1}{P_2} = \frac{\rho_1\ d_1^5\ N_1^3}{\rho_2\ d_2^5\ N_2^3} \tag{5-34}$$

d) Conditions for Rotational speed:

From equation 5-24 we have:

$$\frac{N_1}{N_2} = \frac{\gamma_1\ d_1^2}{\gamma_2\ d_2^2} \tag{5-35}$$

e) Conditions for specific speed:

Taking equations 5-32 and 5-33

$$\frac{Q_1}{Q_2} = \frac{d_1^3\ N_1}{d_2^3\ N_2} \tag{5-36}$$

and

$$\frac{H_1}{H_2} = \frac{d_1^2\ N_1^2}{d_2^2\ N_2^2} \tag{5-37}$$

If the diameter values are eliminated, equations 5-38 and 5-39 are established which define specific speed.

$$\frac{N_1\sqrt{Q_1}}{H_1^{3/4}} = \frac{N_2\sqrt{Q_2}}{H_2^{3/4}} \quad (5\text{-}38)$$

or

$$(Ns)_1 = (N_s)_2 \quad (5\text{-}39)$$

Theoretical Horsepower Determination

An illustration of the determination of power required when certain conditions are given is shown:

Assuming that a flow of 1,000,000 GPM is desired to be moved from near the surface of a 100 ft deep lake to the bottom, and using a pump of: a) 20 ft propeller diameter, b) 30 ft propeller diameter and c) 40 ft propeller diameter and assuming a pump efficiency of 70%.

1. Determination of entrance velocity, V_1

Propeller Diameter	Flow cfs	Area ft^2	Velocity fps
20.0	2228.0	314.15	7.092
30.0	2228.0	706.85	3.152
40.0	2228.0	1256.63	1.773

2. Determination of losses through the pump, H_T

a) Losses due to entrance, H_1

From Hamilton's results (22) values of K_1 are taken as 0.10, thus

$$H_1 = \frac{0.10\ (V_1)^2}{2g} \quad (5\text{-}40)$$

b) Losses due to enlargement, H_2

From Vennard (24), values of K_2 are determined from Table VIII for the diffuser angles and used in the equation:

$$H_2 = K_2 \frac{(V_1 - V_2)^2}{2g} \qquad (5\text{-}41)$$

c) Losses due to exit, H_3

For this study the K_3 is assumed to be unity, then

$$H_3 = (1.0) \frac{(V_1)^2}{2g} \qquad (5\text{-}42)$$

d) Total head losses, H_T

The total head losses, or total dynamic head, TDH, corresponds to the summation of H_1, H_2 and H_3. The TDH, affects the power requirements of the pump, and can be used to determine the efficiency of the pump.

e) Horsepower calculations

Various outlet diameters for each of the three throat diameters were used. The horsepower was calculated for each diameter using an assumed pump efficiency of 0.70.

The minimum horsepower for the optimum outlet diameter is shown in Table VIII.

Values of power for the three diameters of propeller, plotted in Figure 35, show the theoretical horsepower required for a one million gallons per minute (4420 Ac Ft/Day) lake destratifier having a diffuser length of 100 ft. The above relationship is valid for propeller diameters between 20 ft and 40 ft.

TABLE VIII

HORSEPOWER REQUIRED FOR OPTIMUM OUTLET DIAMETER AND A DIFFUSER LENGTH OF 100 FEET

Diameter Propeller	Length of Diffuser	Most Efficient Outlet Diameter	Angle Degree	TDH Ft	Horsepower Hp
20.0	100.0	38.0	10	0.1581	57.0
30.0	100.0	52.0	12	0.0391	14.1
40.0	100.0	64.0	14	0.0139	5.05

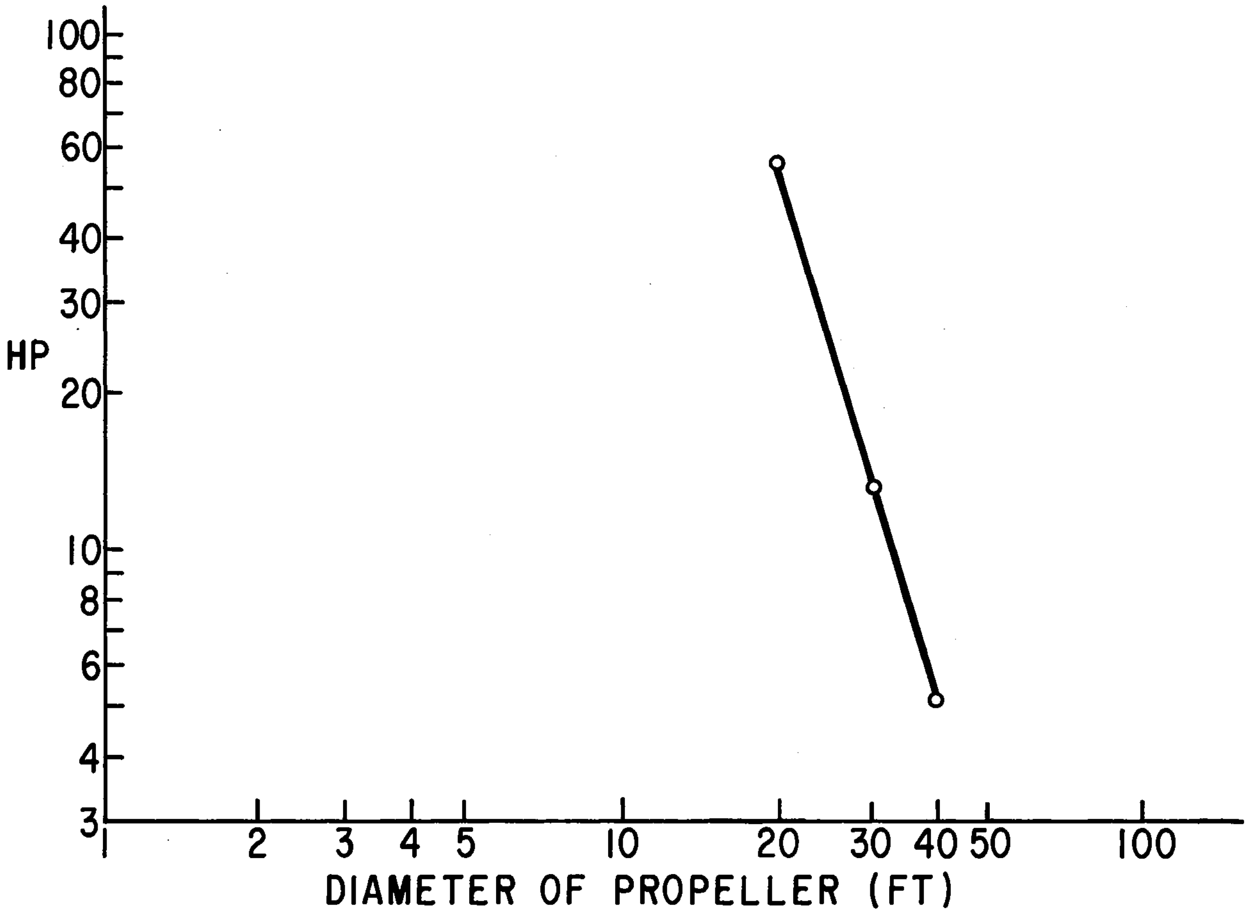

Figure 35. Theoretical Horsepower (70% Eff.) for a 1,000,000 GPM (4420 Ac.Ft./Day) Lake Destratifier

CHAPTER VI

SUMMARY AND CONCLUSIONS

Summary

The objectives of this study were to: 1) Develop a device that will pump large amounts of flow with low input of energy, 2) Establish the relationships among flow rate through the pump and rpm, diffuser outlet diameter, and diffuser length, 3) Determine the relationships of horsepower to rpm, as other variables were varied, 4) Calculate the expected head loss through the device, based on available coefficients and measured velocities, 5) Make estimations of pump efficiency from measured values of horsepower and flow rate using calculated values of head, 6) Establish dimensionless parameters and relationships to assist in the design of larger models.

To accomplish this, a low energy lake destratifier was designed, built and tested. Basically, it consisted of an axial flow type pump with a propeller that moved the water downward, from the surface to the bottom of a body of water.

Four lengths of diffuser (0, 8, 16, and 24 ft.), four diffuser outlet diameters (5, 6, 7, and 8 ft.), and four typical propeller shaft speeds (40, 56, 60, and 80 rpm), were investigated.

Measured values were:

1. Velocity at the throat of the pump, fps.

2. Rpm of the propeller shaft.

3. Power input, watts.

By using multiple regression analysis, prediction equations for the flow through the pump, (Q), as well as power input required, (Ps), were obtained as functions of length of diffuser, outlet diameter of diffuser and propeller rpm. The experimental equations were developed over the range of propeller shaft velocity between 40 and 80 RPM.

Pump entrance losses, expansion losses and exit losses were calculated to determine the Total Dynamic Head, TDH, which would reflect the total power requirements of the pump.

The measured flow and power input values plus the calculated value of the total dynamic head were used to determine the pump efficiency. Observed flow (Q) versus observed propeller shaft speed (RMP) were used to study flow characteristics of the pump. Likewise, observed power input (Ps) versus propeller shaft speed (RPM) were used to study the power input characteristics of the pump.

Conclusions

The following conclusions are made based on an interpretation of the experimental results:

1. The pump was able to pump large flows of water with a low input of energy. A maximum flow of 10,700 GPM or 23.85 CFS was obtained when using 0.498 horsepower. The maximum calculated value of TDH was 0.05 foot.

2. Prediction equation (6-1) was developed to describe the flow (Q) versus rpm, diameter and length for the range of variables studied:

$$Q = -2.07001404 + 0.27627960\ N + 0.54165381\ Do + 0.00920268\ L \qquad (6\text{-}1)$$

The significant factor affecting the flow (Q) of the pump was the rotative velocity of the propeller shaft. The other factors had a minor influence.

3. Prediction equation (6-2) was developed to describe the power (Ps) versus rpm, diameter and length for the range of variables studied:

$$Ps = -0.02433906 + 0.0000012\, N^3 - 0.00112998\, Do - 0.00047625\, L \qquad (6\text{-}2)$$

The significant factor affecting the power input (Ps) of the pump was the rotative velocity of the propeller shaft. The other factors had a minor influence.

4. The efficiencies of the pump were low. The best conditions were found when the pump was operating with a diffuser length of 8.0 ft., a diffuser outlet diameter of 8.0 ft. and using the No. 40 pulley that corresponded to 76.3 rpm. The calculated efficiency for these conditions was 45.85%.

It is believed that low efficiencies were due to: a) propeller inefficiency, b) vane inefficiency and c) diffuser inefficiency. It is possible that low efficiency could be improved by using a more efficient propeller. The selection of a ship screw or a propeller for agitation and stirring of liquids would be indicated. The construction of exit vanes to direct the exit flow and avoid excessive turbulence is suggested. Curved entrance vanes instead of the straight vanes used would probably improve performance. The use of a rigid diffuser may avoid the problems encountered with the flexible diffuser that was used.

5. Calculated values of specific speed varied from 72,000 to 125,000 as compared to values of 7,500 to 14,000 for typical axial

flow pumps.

6. Dimensionless parameters were determined to assist in the design and development of larger models. A 40-foot pump with a capacity of one million gallons per minute would probably require about 10 horsepower, assuming an efficiency of 0.35.

7. The pump should be a practical means of pumping water from the top of a reservoir to the bottom with a low input of energy. This might be a means of destratifying reservoirs, of raising the oxygen content locally near domestic water releases, or of significantly raising the released water temperature for reservoirs with deep intakes to the release system.

Recommendations for Further Study

1. A study to find a more adequate method of diffusing the flow should be conducted. This may include the construction of exit vanes and the use of a rigid diffuser.

2. The effect of smaller propeller shaft speed on pump flow and power requirements should be considered.

3. A study to relate the flow of the pump to the area and volume of the lake to determine its area of influence should be conducted.

4. A study to utilize wind power so the pump is not dependent on an external source of power is needed.

5. A study to determine the effects of operation of the pump on the temperature and dissolved oxygen distribution in a lake, before and after stratification is needed.

6. Construction of a larger scale pump to determine its operating characteristics will be needed if the device is to be applied to larger lakes.

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APPENDIX A

CALCULATED VALUES OF HEAD LOSS, EFFICIENCY, HORSEPOWER AND SPECIFIC SPEED FOR EACH TEST

TABLE IX

CALCULATED VALUES OF LOSSES FOR EACH CONDITION OF FLOW DURING THE EXPERIMENT

Diffuser Length (Ft)	Diffuser Outlet Diameter (Ft)	Velocity at Throat (FPS)	Pulley Size No.	Loss at Entrance H1 (Ft)	Loss at Enlargement H2 (Ft)	Loss at Exit H3 (Ft)	Total Head Loss HT (Ft)
8.00	5.00	1.335	20.	0.0028	0.0018	0.0066	0.0112
8.00	5.00	1.756	28.	0.0048	0.0031	0.0115	0.0194
8.00	5.00	1.883	30.	0.0055	0.0036	0.0132	0.0223
8.00	5.00	2.320	40.	0.0084	0.0054	0.0201	0.0339
8.00	6.00	1.330	20.	0.0027	0.0043	0.0032	0.0102
8.00	6.00	1.792	28.	0.0050	0.0078	0.0058	0.0186
8.00	6.00	1.896	30.	0.0056	0.0087	0.0065	0.0208
8.00	6.00	2.420	40.	0.0091	0.0142	0.0105	0.0339
8.00	7.00	1.330	20.	0.0027	0.0073	0.0017	0.0117
8.00	7.00	1.823	28.	0.0052	0.0136	0.0032	0.0220
8.00	7.00	1.935	30.	0.0058	0.0154	0.0036	0.0248
8.00	7.00	2.400	40.	0.0089	0.0236	0.0056	0.0382
8.00	8.00	1.367	20.	0.0029	0.0114	0.0011	0.0153
8.00	8.00	1.853	28.	0.0053	0.0209	0.0020	0.0282
8.00	8.00	1.975	30.	0.0061	0.0238	0.0022	0.0320
8.00	8.00	2.473	40.	0.0095	0.0373	0.0035	0.0502

TABLE IX (Continued)

Diffuser Length (Ft)	Diffuser Outlet Diameter (Ft)	Velocity at Throat (FPS)	Pulley Size No.	Loss at Entrance H1 (Ft)	Loss at Enlargement H2 (Ft)	Loss at Exit H3 (Ft)	Total Head Loss HT (Ft)
16.00	5.00	1.345	20.	0.0028	0.0013	0.0067	0.0109
16.00	5.00	1.800	28.	0.0050	0.0024	0.0121	0.0195
16.00	5.00	1.930	30.	0.0058	0.0027	0.0139	0.0224
16.00	5.00	2.325	40.	0.0084	0.0039	0.0202	0.0325
16.00	6.00	1.350	20.	0.0028	0.0027	0.0033	0.0088
16.00	6.00	1.800	28.	0.0050	0.0048	0.0058	0.0157
16.00	6.00	1.940	30.	0.0058	0.0056	0.0068	0.0182
16.00	6.00	2.460	40.	0.0094	0.0090	0.0109	0.0293
16.00	7.00	1.340	20.	0.0028	0.0044	0.0017	0.0089
16.00	7.00	1.840	28.	0.0053	0.0083	0.0033	0.0168
16.00	7.00	1.960	30.	0.0060	0.0094	0.0037	0.0191
16.00	7.00	2.450	40.	0.0093	0.0147	0.0058	0.0298
16.00	8.00	1.380	20.	0.0030	0.0064	0.0011	0.0104
16.00	8.00	1.890	28.	0.0055	0.0120	0.0020	0.0195
16.00	8.00	1.970	30.	0.0060	0.0130	0.0022	0.0212
16.00	8.00	2.480	40.	0.0096	0.0206	0.0035	0.0337

TABLE IX (Continued)

Diffuser Length (Ft)	Diffuser Outlet Diameter (Ft)	Velocity at Throat (FPS)	Pulley Size No.	Loss at Entrance H1 (Ft)	Loss at Enlargement H2 (Ft)	Loss at Exit H3 (Ft)	Total Head Loss HT (Ft)
24.00	5.00	1.295	20.	0.0026	0.0009	0.0063	0.0098
24.00	5.00	1.749	28.	0.0047	0.0017	0.0114	0.0179
24.00	5.00	1.853	30.	0.0053	0.0019	0.0128	0.0201
24.00	5.00	2.308	40.	0.0083	0.0030	0.0199	0.0311
24.00	6.00	1.320	20.	0.0027	0.0021	0.0031	0.0080
24.00	6.00	1.790	28.	0.0050	0.0039	0.0058	0.0146
24.00	6.00	1.870	30.	0.0054	0.0043	0.0063	0.0160
24.00	6.00	2.320	40.	0.0084	0.0065	0.0097	0.0246
24.00	7.00	1.371	20.	0.0029	0.0036	0.0018	0.0084
24.00	7.00	1.762	28.	0.0048	0.0060	0.0030	0.0138
24.00	7.00	1.864	30.	0.0054	0.0067	0.0034	0.0154
24.00	7.00	2.372	40.	0.0087	0.0108	0.0055	0.0250
24.00	8.00	1.345	20.	0.0028	0.0046	0.0010	0.0084
24.00	8.00	1.846	28.	0.0053	0.0086	0.0019	0.0159
24.00	8.00	1.940	30.	0.0058	0.0096	0.0021	0.0175
24.00	8.00	2.390	40.	0.0089	0.0145	0.0032	0.0266

TABLE X

PUMP EFFICIENCIES FOR CALCULATED VALUES OF HEAD AND MEASURED VALUES OF HORSEPOWER AND FLOW THROUGH THE PUMP

Diffused Length (Ft)	Outlet Diameter (Ft)	Pulley Size* No.	Measured Flow (CFS)	Calculated Total Head Loss (Ft)	Efficiency of Pump Percent	Horsepower (HP)
8.00	5.00	20.	12.8	0.01121	40.9	0.040
8.00	5.00	28.	16.9	0.01940	22.6	0.165
8.00	5.00	30.	18.1	0.02231	20.9	0.220
8.00	5.00	40.	22.3	0.03386	17.6	0.488
8.00	6.00	20.	12.8	0.01023	37.2	0.040
8.00	6.00	28.	17.2	0.01857	20.4	0.178
8.00	6.00	30.	18.2	0.02079	19.6	0.220
8.00	6.00	40.	23.3	0.03387	18.0	0.498
8.00	7.00	20.	12.8	0.01173	38.7	0.044
8.00	7.00	28.	17.5	0.02203	26.1	0.168
8.00	7.00	30.	18.6	0.02482	25.0	0.210
8.00	7.00	40.	23.1	0.03818	20.9	0.480
8.00	8.00	20.	13.1	0.01535	45.9	0.050
8.00	8.00	28.	17.8	0.02820	34.6	0.165
8.00	8.00	30.	19.0	0.03204	31.4	0.220
8.00	8.00	40.	23.8	0.05023	27.7	0.490

TABLE X (Continued)

Diffused Length (Ft)	Outlet Diameter (Ft)	Pulley Size* No.	Measured Flow (CFS)	Calculated Total Head Loss (Ft)	Efficiency of Pump Percent	Horsepower (HP)
16.00	5.00	20.	12.9	0.01087	29.0	0.055
16.00	5.00	28.	17.3	0.01947	20.2	0.190
16.00	5.00	30.	18.6	0.02238	20.5	0.230
16.00	5.00	40.	22.4	0.03248	15.3	0.540
16.00	6.00	20.	13.0	0.00882	26.0	0.050
16.00	6.00	28.	17.3	0.01567	16.7	0.185
16.00	6.00	30.	18.7	0.01821	16.8	0.230
16.00	6.00	40.	23.7	0.02928	15.3	0.515
16.00	7.00	20.	12.9	0.00892	26.1	0.050
16.00	7.00	28.	17.7	0.01682	18.3	0.185
16.00	7.00	30.	18.8	0.01909	18.2	0.225
16.00	7.00	40.	23.6	0.02983	15.7	0.510
16.00	8.00	20.	13.3	0.01042	41.4	0.038
16.00	8.00	28.	18.2	0.01955	23.1	0.175
16.00	8.00	30.	18.9	0.02124	20.8	0.220
16.00	8.00	40.	23.8	0.03366	18.3	0.498

TABLE X (Continued)

Diffused Length (Ft)	Outlet Diameter (Ft)	Pulley Size* No.	Measured Flow (CFS)	Calculated Total Head Loss (Ft)	Efficiency of Pump Percent	Horsepower (HP)
24.00	5.00	20.	12.5	0.00980	30.8	0.045
24.00	5.00	28.	16.8	0.01788	18.2	0.188
24.00	5.00	30.	17.8	0.02007	18.5	0.220
24.00	5.00	40.	22.2	0.03114	15.2	0.518
24.00	6.00	20.	12.7	0.00796	28.7	0.040
24.00	6.00	28.	17.2	0.01463	15.9	0.180
24.00	6.00	30.	18.0	0.01597	14.8	0.220
24.00	6.00	40.	22.3	0.02458	12.3	0.505
24.00	7.00	20.	13.2	0.00835	25.0	0.050
24.00	7.00	28.	16.9	0.01380	14.8	0.180
24.00	7.00	30.	17.9	0.01544	14.0	0.225
24.00	7.00	40.	22.8	0.02501	12.8	0.507
24.00	8.00	20.	12.9	0.00843	41.3	0.030
24.00	8.00	28.	17.8	0.01588	19.1	0.168
24.00	8.00	30.	18.7	0.01754	18.1	0.205
24.00	8.00	40.	23.0	0.02662	13.8	0.505

*Nominal RPM for Propeller = 2 * Pulley Size

TABLE XI

CALCULATED SPECIFIC SPEED OF THE PUMP FOR EACH CONDITION OF THE EXPERIMENT

Outlet Diameter Ft	Diffuser Length ft	Pulley No. 20	Pulley No. 28	Pulley No. 30	Pulley No. 40
5.0	8.0	90324.19	92133.75	94025.94	100429.80
6.0	8.0	95337.00	97973.81	99455.88	100429.80
7.0	8.0	86475.00	87754.13	87219.94	89280.00
8.0	8.0	72220.44	73511.50	74176.56	74789.06
5.0	16.0	93975.44	96553.19	97873.13	101252.20
6.0	16.0	108769.60	110328.30	111135.40	111825.30
7.0	16.0	106039.30	106927.30	107813.00	108558.00
8.0	16.0	97579.44	98631.63	99781.13	100446.30
5.0	24.0	99619.69	101421.00	102687.40	104118.10
6.0	24.0	118353.80	120346.20	122256.40	123799.10
7.0	24.0	115294.10	121677.30	122717.10	125620.30
8.0	24.0	112995.00	114473.60	115971.10	117576.80

APPENDIX B

ORIGINAL PUMP DATA FROM EXPERIMENTS

TABLE XII

PUMP DATA FROM EXPERIMENTS

Experiment No.	Diffuser Diameter Ft	Diffuser Length Ft	Pulley Size No.	Velocity at Throat Fps	Flow of Pump Cfs.	Propeller Shaft Rpm	Registered Power Hp.
1	3.5	0.0	20	1.16	11.160	40.74	0.065
2	3.5	0.0	28	1.50	14.431	56.57	0.175
3	3.5	0.0	30	1.58	15.201	60.20	0.22
4	3.5	0.0	40	1.85	17.799	76.81	0.510
5	5.0	8.0	20	1.33	12.844	41.00	0.054
6	5.0	8.0	28	1.76	16.900	55.01	0.185
7	5.0	8.0	30	1.88	18.119	60.20	0.220
8	5.0	8.0	40	2.32	22.321	75.77	0.488
9	6.0	8.0	20	1.33	12.796	40.48	0.048
10	6.0	8.0	28	1.79	17.241	56.05	0.185
11	6.0	8.0	30	1.89	18.241	60.20	0.220
12	6.0	8.0	40	2.42	23.287	77.59	0.521
13	7.0	8.0	20	1.33	12.796	40.67	0.040

TABLE XII (Continued)

Experiment No.	Diffuser Diameter Ft	Diffuser Length Ft	Pulley Size No.	Velocity at Throat Eps	Flow of Pump Cfs.	Propeller Shaft Rpm	Registered Power Hp.
14	7.0	8.0	28	1.82	17.542	56.57	0.185
15	7.0	8.0	30	1.94	18.617	59.68	0.220
16	7.0	8.0	40	2.40	23.090	75.77	0.480
17	8.0	8.0	20	1.37	13.152	41.00	0.040
18	8.0	8.0	28	1.85	17.828	56.57	0.184
19	8.0	8.0	30	1.98	19.001	60.84	0.220
20	8.0	8.0	40	2.47	23.796	76.81	0.510
21	5.0	16.0	20	1.34	12.940	41.52	0.060
22	5.0	16.0	28	1.80	17.318	57.09	0.190
23	5.0	16.0	30	1.93	18.569	62.05	0.230
24	5.0	16.0	40	2.32	22.369	77.33	0.518
25	6.0	16.0	20	1.35	12.988	41.00	0.040
26	6.0	16.0	28	1.80	17.318	55.45	0.185

TABLE XII (Continued)

Experiment No.	Diffuser Diameter Ft	Diffuser Length Ft	Pulley Size No.	Velocity at Throat Eps	Flow of Pump Cfs.	Propeller Shaft Rpm	Registered Power Hp.
27	6.0	16.0	30	1.94	18.664	60.20	0.230
28	6.0	16.0	40	2.46	23.668	76.81	0.515
29	7.0	16.0	20	1.34	12.892	40.48	0.050
30	7.0	16.0	28	1.84	17.703	56.05	0.185
31	7.0	16.0	30	1.96	18.857	60.20	0.225
32	7.0	16.0	40	2.45	23.571	75.77	0.510
33	8.0	16.0	20	1.38	13.277	41.24	0.038
34	8.0	16.0	28	1.89	18.184	57.09	0.175
35	8.0	16.0	30	1.97	18.953	60.20	0.220
36	8.0	16.0	40	2.48	23.860	76.29	0.498
37	5.0	24.0	20	1.29	12.459	41.52	0.045
38	5.0	24.0	28	1.75	16.827	57.09	0.188
39	5.0	24.0	30	1.85	17.828	61.24	0.220

TABLE XII (Continued)

Experiment No.	Diffuser Diameter Ft	Diffuser Length Ft	Pulley Size No.	Velocity at Throat Eps	Flow of Pump Cfs.	Propeller Shaft Rpm	Registered Power Hp.
40	5.0	24.0	40	2.31	22.205	77.34	0.518
41	6.0	24.0	20	1.32	12.700	41.78	0.040
42	6.0	24.0	28	1.79	17.222	57.61	0.180
43	6.0	24.0	30	1.87	17.991	61.14	0.220
44	6.0	24.0	40	2.32	22.321	76.81	0.505
45	7.00	24.0	20	1.32	12.709	41.42	0.050
46	7.00	24.0	20	1.76	16.952	56.18	0.180
47	7.00	24.0	30	1.86	17.929	59.94	0.225
48	7.00	24.0	40	2.37	22.821	78.08	0.507
49	8.0	24.0	20	1.34	12.940	41.26	0.030
50	8.0	24.0	28	1.85	17.760	57.37	0.168
51	8.0	24.0	30	1.94	18.665	61.08	0.205
52	8.0	24.0	40	2.39	22.994	76.29	0.505

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VITA

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